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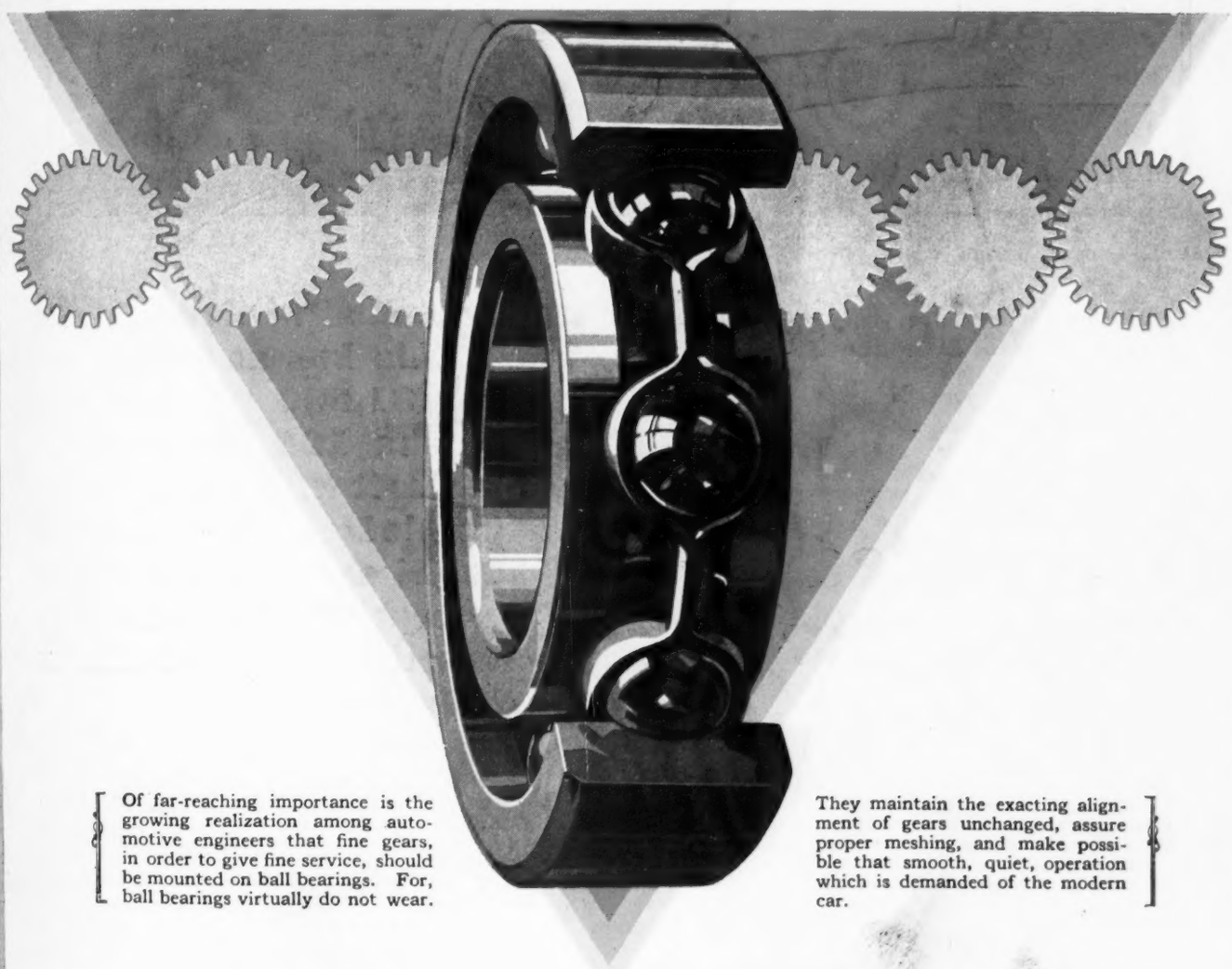

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TORSIONAL

Vibration of Crankshafts —Its Causes and Control

Lower harmonics of gas pressure curve are principal factor—not inertia force

By P. M. Heldt

EVER since six-cylinder cars came into extensive use there has been considerable trouble from "roughness" of the engines within certain speed ranges. The engines were said to have "periods," and these in many cases were decidedly difficult to deal with.

Increase in engine speeds accentuated the difficulties, and it is safe to say that there is not a single concern manufacturing six or eight-cylinder vertical engines of modern (high speed) type that has not been bothered by them. Much research work has been done of late to elucidate the problem, and engineers no longer need to grapple with it entirely in the dark, as was the case up to about ten years ago.

We now know that these "rough spots" or periods in the speed range of the engine are due to synchronism between the natural period of vibration of the crank-assembly and some component (or harmonic) of the crank moment.

In Fig. 1 is shown a crank moment or torque diagram of a six-cylinder engine. It will be seen that the moment varies within wide limits. Every time the crank moment increases it tends to "wind up" the crankshaft as it were. A six throw crankshaft is comparatively flexible, and its rear end naturally is restrained by the inertia of the flywheel and the torque load on the engine. Therefore, every time the turning moment increases the shaft will be slightly "wound up," and immediately thereafter, when the turning moment decreases again, it will unwind. This effect, which is known as forced torsional vibration, occurs at all engine speeds, but at other than certain critical speeds —the periods—it does not give rise to any difficulties in operation.

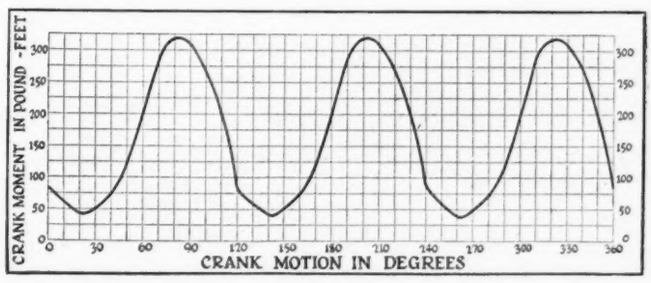


Fig. 1. *Crank moment diagram of six-cylinder engine*

The amplitude of this forced vibration, which is directly proportional to the range in torque fluctuation and inversely proportional to the torsional rigidity of the crankshaft, is very small, because of the large diameters commonly employed for the crankshafts of high speed engines. The

period of forced vibrations in a six-cylinder engine—in other words, the time of a complete cycle—is equal to that of 120 deg. of crank motion.

The crankshaft is capable of vibrating as a torsional pendulum and, like every pendulum, has a definite frequency of vibration. For instance, if we hold the flywheel rigidly and then twist the forward end around the axis of the crank, upon releasing it the forward end will swing back and forth to opposite sides of its normal position with gradually decreasing amplitude, coming to rest after a brief period as a result of the damping action, which is due to air friction and in part also to molecular friction.

Laws of the Pendulum

We have here all of the factors which are involved in the operation of a simple pendulum. In the first place, a force must be applied to move the pendulum out of its position of equilibrium or its position of rest. This is known as the exciting force. Application of this force causes a certain amount of energy to be stored up in the moving parts of the pendulum, and during each complete cycle this energy is twice converted from one form to another. When the pendulum swings through its position of equilibrium its parts move at their maximum speed, and all of the energy then is in the form of kinetic energy. At each end of the swing the parts are momentarily at rest and possess no kinetic energy whatever. All of the energy has been converted into the static or potential form. In the ordinary pendulum the weight has been lifted a certain distance against the force of gravity and is capable of doing mechanical work while returning to its lowest position, that of equilibrium. In a torsional pendulum the fibers of the wire are subjected to torsion and have energy stored up in them in the form of molecular stress.

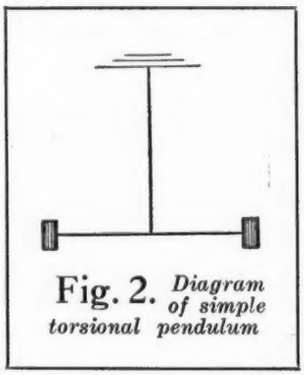


Fig. 2. *Diagram of simple torsional pendulum*

In order to readily understand the laws of the torsional pendulum, one must assume an equivalent of the "simple pendulum" of physics text books. Such a simple torsional pendulum is represented in Fig. 2. A steel wire is rigidly held at its upper end, and at its lower end it carries a horizontal bar with weights at its opposite ends. The wire has a certain torsional rigidity, but its mass is assumed to be negligible. The cross-bar is assumed to be absolutely rigid and also to have negligible mass. If the cross-bar with the weights is turned through an angle around the axis of the wire and then released, it will oscillate around its normal position, or position of rest, and the rate of oscillation will be independent of the amplitude. The damping force will gradually reduce the amplitude of swing but the number of swings per unit of time will remain the same. The number of complete oscillations or cycles per minute of such a pendulum is given by the equation—

$$N = \frac{30}{\pi}\sqrt{\frac{J\,E_t}{L\,R^2\,M}}$$

where J is the polar moment of inertia of the wire.

E_t, the coefficient of torsional elasticity of steel (about 12,000,000 lb. p. sq. in.)

R, the least radius of gyration of the revolving masses or merely the radius if all the masses are considered concentrated at a definite distance from the axis.

M, the sum of the two masses.

Of the above factors, J, E_t and L determine the torsional stiffness of the wire, while M and R^2 determine the moment of inertia of the oscillating mass. The rate of oscillation varies directly as the square root of the torsional stiffness of the wire and inversely as the square root of the moment of inertia of the oscillating mass.

The polar moment of inertia of a solid round shaft is equal to $\pi\ d^4/32$, and substituting this for J in the above equation, inserting the value of the torsional coefficient of elasticity (12,000,000) and combining all of the known values into a single one, we may write the equation for the number of cycles per minute in the following form:

$$N = 58{,}800\ \sqrt{\frac{d^4}{L}}\ \sqrt{\frac{1}{R^2\,W}}$$

Of the two radical terms the first one represents the torsional stiffness of the shaft and the second the reciprocal of the inertia of the oscillating masses.

The torsional vibration of the crankshaft of an automobile engine is somewhat different from that of the simple model illustrated in Fig. 2, inasmuch as no part of the crankshaft is rigidly held. In this case the fixed end is replaced by a nodal point generally located a short distance ahead of the flywheel. The part of the crankshaft marked by this point rotates at uniform angular velocity, whereas when there is torsional vibration that portion of the crankshaft forward of the nodal point has a periodic motion of sinusoidal form super-imposed upon its uniform rotary motion, while the part back of the nodal point has imposed upon it a periodic motion of the same frequency as that of the forward section but out of phase with it by 180 deg. That is, as the forward end of the crankshaft snaps ahead, the rear end and the flywheel slow up in speed or oscillate toward the opposite side of their position of equilibrium. The location of the nodal point is determined by the fact that the products of the moments of inertia of the two sections into their respective angular velocities of swing must be equal.

The Torsional Pendulum

A model of a torsional pendulum with a single nodal point is shown in Fig. 3. Assuming the shaft to be of constant diameter, the angular motions of its different sections are directly proportional to their distance from the nodal point. The inertia masses of the crank-arms and crank-pins are assumed to be concentrated at one point and to have a polar moment of inertia J_1, while the flywheel has a polar moment of inertia J_2. Therefore denoting the angular velocities of the front and rear ends by ω_1 and ω_2, respectively, we have

$$J_1\,\omega_1 = J_2\,\omega_2$$

$$c\,l_1\,J_1 = c\,l_2\,J_2 = c\,(L - l_1)\,J_2$$

$$= c\,L\,J_2 - c\,l_1\,J_2$$

$$l_1 = \frac{J_2}{J_2 + J_1}\,L$$

Hence, if J_1 is very small as compared with J_2, l_1 will be nearly equal to L; that is, the nodal point will be very close to the flywheel. On the other hand, if J_1 and J_2 were equal, then the nodal point would be midway between the center of gravity of the flywheel and the center of gravity of the crank throws. Thus it is possible to move the nodal point along the axis of the shaft by varying the moment of inertia of the flywheel or that of the crank throws. For instance, adding balance weights to a crankshaft will move the nodal point forward, and so will the addition of a front or center flywheel.

The formula for the number of cycles per minute given in the foregoing can be applied directly to a torsionally vibrating system with a single node by measuring the length L_1 from the nodal point.

When a torsional pendulum is merely given a single impulse and is then left to itself, it soon comes to rest through the action of the damping force. By damping force must be understood the sum of all those resistances to the torsional vibration which when overcome result in the conversion of mechanical into heat energy and the dissipation of the latter.

On the other hand, if the pendulum receives an impulse at intervals equal to its natural period, then the amplitude of its vibratory motion increases until a sort of equilibrium is established between the exciting force and the damping force, when exactly the same amount of energy is dissipated in overcoming the damping force as is imparted to the pendulum by the exciting force. Under these conditions, that is when there is a periodic exciting force which is of the same frequency as the natural vibration of the shaft, there is resonance, and the shaft is said to run at a critical speed.

Harmonics of Crank Moment

The crank moment of any multi-cylinder engine is quite variable. The two types of engine of particular interest in this connection are the six-cylinder and the eight-cylinder vertical. In both of these the crank moment is always positive but it varies through a considerable range. Such a periodically variable crank moment or torque can be resolved into a mean continuous crank moment and an alternating periodic crank moment which, according to Fourier's theorem, can be resolved into a number of sine-wave-form or harmonic moments, the periods of which bear to each other the relations 1/1: 1/2: 1/3: ¼, etc. That is to say, there will be one sine-wave-form or harmonic moment the period of which is, say, t; then there will be another sine wave force with a period $t/2$, another with a period $t/3$ and so on. In the case of the crank moment of a multi-cylinder engine the period or time of one complete cycle of the primary harmonic force is equal to the time between successive explosions; that is for a six-cylinder engine it is equal to the time of one-third a crankshaft revolution. The frequencies of the different harmonics, of course, bear to each other the reverse relation as the periods. Thus if the frequency of the first harmonic is N, that of the second will be $2N$, that of the third $3N$, and so on.

In a single-cylinder engine the crank moment depends upon both the gas pressure and the inertia forces, and this applies also to the moment on the individual throws of a multi-cylinder crank. The inertia forces are harmonic forces with periods bearing to each other the relation 1/1, 1/2, 1/3, etc., and one therefore might expect these forces to be responsible for torsional vibration. It can be readily shown, however, that in six and eight-cylinder vertical engines at least the primary and secondary inertia forces do not act as exciting forces for torsional vibration.

In both types of engine, pairs of cylinders located symmetrically with respect to the transverse center plane of the engine are in phase, and the resultant of any harmonic due to the inertia forces on the reciprocating parts of such a pair of cylinders acts in this central plane. The resultant from a single pair of cylinders, say the foremost and rearmost one, therefore would act on a length of shaft equal to the distance from the central plane to the nodal point. However, in a six-cylinder engine the resultants in the central plane from the three pairs of cylinders, and in an eight-cylinder the resultants from the four pairs, cancel each other out for both the primary and secondary forces. Therefore, it can be stated that in a six-cylinder and an eight-cylinder vertical engine with the conventional crank arrangements neither the primary nor the secondary unbalanced force has any part in the production of torsional vibration and that at the highest critical speed within the working range this vibration is due entirely to the gas pressure. This has been verified, moreover, by tests in which an engine was turned over by belt or an electric motor, at a speed at which there was a greal deal of "roughness" when it was running under its own power. When thus operated the roughness is entirely absent.

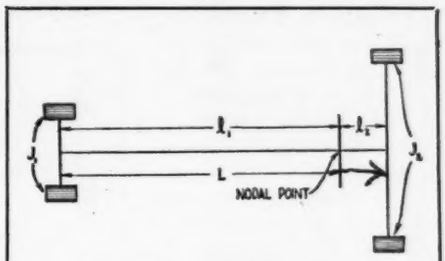


Fig. 3. *Diagram of torsional pendulum with single nodal point*

Any harmonic or sinusoidal force will produce a forced vibration of the crankshaft at all speeds, but it will produce vibration of considerable amplitude only when it is entirely or nearly synchronous with the natural period of vibration of the shaft. When of a frequency differing materially from that of the natural vibration of the shaft its effect will be negligible.

According to the researches of C. E. Summers, of the General Motors Research Corp., six-cylinder crankshafts of high speed automobile engines have a natural period of oscillation of 180 to 270 complete cycles per second or 10,800-16,200 per minute. The first harmonic of the torque curve passes through three complete cycles per revolution, hence this will be in synchronism with the natural period of the crankshaft in different engines, at speeds between 10,800/3=3,600 and 16,200/3=5,400 r. p. m., which speeds are beyond the operating ranges of all stock passenger car engines.

In analyzing the exciting forces and their effect on the crankshaft it is impractical to start with the crank moment diagram of the complete engine, because this is a composite moment which does not act at any particular point. It is therefore customary to base the analysis upon the gas pressure diagram or indicator diagram and the curve of crank moments resulting therefrom. By methods of harmonic analysis this gas pressure crank moment diagram can be resolved into its various harmonics, and corresponding harmonics of the different cylinders or crank throws can then be combined, taking account of their phase relations. Methods of harmonic analysis have been developed especially in connection with the study of alternating electric currents, and the subject is dealt with in the *Electrical Journal*, 1908, Vol. V, p. 386.

Analysis of Gas Pressure Moment

It appears that the amplitudes of succeeding harmonics due to the gas pressure do not decrease nearly as rapidly as those due to the unbalanced forces, of which the second is equal to about one-fourth of the first. An analysis of the gas pressure torque due to a single cylinder with an ideal form of diagram showing 335 lb. maximum and 103 lb. mean effective pressure, has been made by Chief Engineer Plunzke (*Der Motorwagen*, Feb. 28, 1926), who found that the amplitudes of the first five harmonics are $\pm$ 32.4 lb. p. sq. in., $\pm$ 34 lb. p. sq. in., $\pm$ 31.5 lb. p. sq. in., 25.3 lb. p. sq. in., and 18.6 lb. p. sq. in. It will thus be seen that the amplitude of the second harmonic is even greater than that of the first. Mr. Plunzke analyzed the

first eleven harmonics, and his results are given in Fig. 4. These, it must be remembered, are due to the gas pressures in a single cylinder engine.

The frequency of the first harmonic of the gas pressure torque is equal to one-half the crankshaft revolutions per minute and its period is equal to the time of two revolutions of the crankshaft. In a multi-cylinder engine no harmonic of the gas pressure crank moment ever acts alone on the crankshaft, but it always combines with harmonics of the same order due to the gas pressures in the rest of the cylinders. The fundamental and all of the lower harmonics of the gas pressure crank moment are of too low periodicity to produce synchronous vibration in the crankshaft, and in a six-cylinder engine it is only the sixth and higher harmonics of the gas pressure torque which cause trouble.

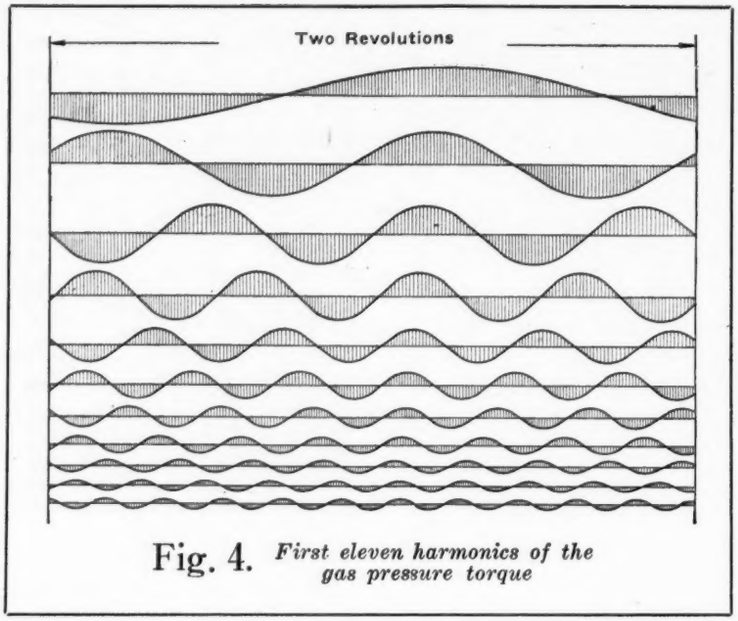


Fig. 4. *First eleven harmonics of the gas pressure torque*

The twisting moments due to the inertia forces are comparatively easy to deal with, for the reason that these forces due to cylinders located symmetrically on opposite sides of the central plane are in phase, and their effects may be represented by a single harmonic acting in the central plane. The gas pressures in cylinders located symmetrically on opposite sides of the central plane are not in phase and this method of treatment therefore is not applicable in their case. The gas pressure in each cylinder produces a series of harmonics which differ in frequency, amplitude and phase.

Phase Equality for All Cranks

Harmonics differing in frequency cannot re-inforce each other, as it were, for though they may act in the same direction for an instant, an instant later they will be in opposition to each other, and it is therefore impossible for them to combine to build up vibration of great amplitude. Corresponding harmonics of the gas pressure moments of the different cylinders are equal in frequency and (theoretically at least) in amplitude, but they usually differ in phase. For instance, the first harmonic due to the gas pressure in the rear cylinder is directly opposed in phase to that due to the gas pressure in the front cylinder, while in a four-cylinder engine the first harmonic of the second and third cylinders also are opposed to each other and differ in phase by 90 deg. from those of the two end cylinders, respectively.

However, in the case of some of the higher harmonics, those of all of the cylinders are in phase. This can be clearly seen from Fig. 5 which in the upper part shows the crank moment due to the gas pressure in one cylinder for more than one complete cycle. The central horizontal line is the line of average crank moment produced by a single cylinder. During most of the power stroke the cylinder develops far more than the average torque, while during the rest of the cycle the torque is below average. Therefore, regarding the total torque as made up of a constant mean value and a periodic value super-imposed thereon, the curve in the upper part of the figure represents this periodic value.

It was stated in the foregoing that such a periodic force can be resolved into a series of harmonic forces of which the fundamental one has the same frequency as the periodic force itself, while the other harmonics have twice, three times, four times, etc., this frequency. In the lower part of the figure the first, second and fourth harmonics of the crank moment curve have been laid out in a rough way. The amplitudes and phase relations of the different harmonics at different equidistant points of the cycle can be found from the values of the crank moment by methods of harmonic analysis. All of the different harmonics are substantially in phase when the crank moment is at its maximum.

Now let us assume we are dealing with a four-cylinder engine with a firing order 1-2-4-3. The four-cylinder example is used here because with it at any particular moment each of the four pistons performs a different stroke of the cycle, and the relations therefore are particularly simple. In the diagram Cylinder No. 1 is on the power stroke and the first harmonic of its crank moment is positive and attains its maximum value during the first half of the stroke. The first harmonic of the crank moment of Cylinder No. 2 at the same time is zero, that of Cylinder No. 3 also is zero, while that of Cylinder No. 4 has its maximum negative value. Thus when crank No. 1 twists the crankshaft in one direction, crank No. 4 tends to twist it in the opposite direction.

The second harmonic is at its maximum positive value simultaneously in Cylinders Nos. 1 and 4 and at its maximum negative value in Cylinders Nos. 2 and 3. The fourth harmonic, it will be noticed, is at its maximum positive value simultaneously in all four cylinders. Therefore, the

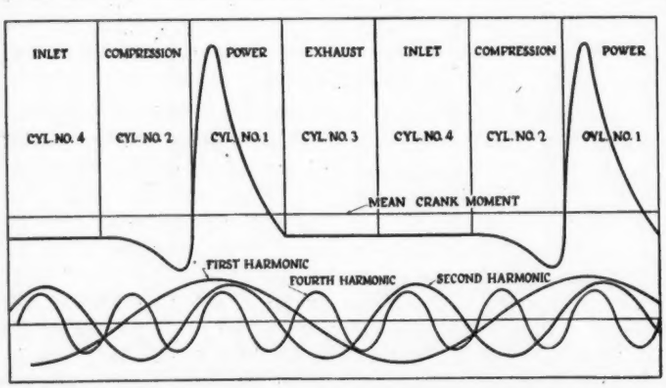


Fig. 5. *Gas pressure torque curve and its first, second and fourth harmonics*

fourth harmonics of all four cranks tend to twist the crankshaft in the same direction. The same holds true of the eight, twelfth and sixteenth harmonics. In the case of a six-cylinder engine it would be the sixth, twelfth, eighteenth, etc.

It should be pointed out that the first, second and fourth harmonics referred to here are those of the crank moment due to the gas pressure in a single cylinder, which, of course, must not be confounded with the different harmonics of the crank moment curve of a multi-cylinder engine. For instance, the period of the first harmonic of the single-cylinder gas pressure crank moment is equal to the time of one engine cycle or two crankshaft revolutions, while the period of the first harmonic of the crank moment curve of a six-cylinder engine is equal to the time of one-third a crankshaft revolution. Hence the first harmonic of a six-cylinder crank moment diagram has the same frequency as the sixth harmonic of the gas pressure torque.

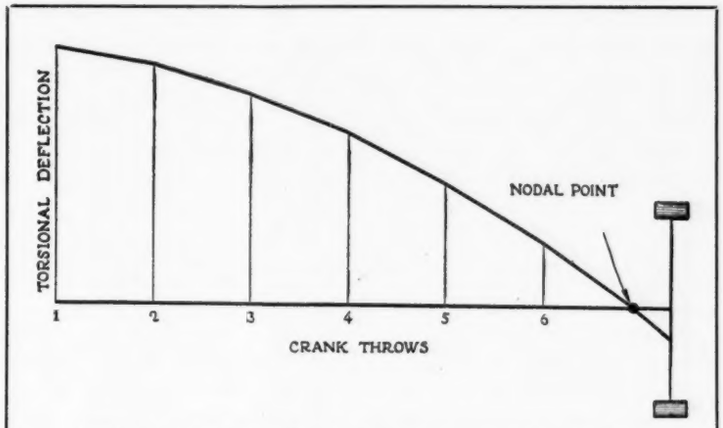


Fig. 6. *"Curve" of flexure of a six cylinder crankshaft under harmonics which are in phase for all crank throws*

If a shaft is held rigidly at one end and is subjected to a twisting force at the other, the twist or torsional displacement along its length will vary in direct proportion to the distance from the fixed end. But if there are twisting forces distributed along the length of the shaft, all acting in the same direction, then the twist at any point will depend upon the number of twisting forces effective at that point and the sum of their magnitudes. For instance, in the case of a six-cylinder engine, where the sixth harmonics are in phase for all of the six cranks, the angular displacement along the length of the crank will be as indicated in Fig. 6.

No attempt will be made in this article to describe methods for predetermining the natural periods of oscillation of crankshaft assemblies. While some fairly simple graphical methods have been worked out, it is the writer's opinion that in general the calculations required are so involved and the assumptions which must be made are so uncertain that no very accurate results can be expected, especially with crank arms of irregular shape as often used with six-cylinder engines. It would seem desirable, however, that some dimensioned lay-outs of crankshaft assemblies with their natural frequencies of torsional vibration should be published, as this would familiarize engineers with the subject and would make it possible to predict the approximate critical speeds of a new shaft by comparison with older ones of which the characteristics are known.

It will be well, in this connection, to take another look at the formula for the rate of vibration of a simple torsional pendulum, viz.,

$$N = 58{,}000 \sqrt{\frac{d^4}{l}} \sqrt{\frac{1}{R^2 W}}$$

This shows that the factor which has the most influence on the natural rate of torsional vibration is the shaft diameter *d*. In fact, this rate, and, hence, the critical speed, increases substantially as the square of this diameter—*substantially* because any increase in shaft diameter will also increase the moment of inertia (R^2W), though not to any great extent.

Lengthening the crankshaft lowers the natural rate of vibration, and hence the critical speed, but the effect is not so pronounced, as the number of cycles per minute varies only as the square root of the length. In this case, too, the effect is slightly modified by the fact that any increase in length entails an increase in weight. But since the additional weight is all close to the axis—assuming the additional length to be in the main bearings—its effect on the moment of inertia is slight.

The least radius of gyration has a quite important influence and, in fact, the "rate of vibration" is inversely proportional to it. This radius *R* is dependent upon the crank radius, and any increase in crank throw would cause *R* to increase in substantially the same proportion, at least if the crankshaft were a simple one and there were no extraneous inertia masses of a kind which are not affected by a change in crank throw. Hence, if it is desired to raise the critical speeds, shortening the stroke is a very effective procedure.

Influence of Weight

Finally, the weight *W* has an influence— the greater the weight, the lower the critical speed. This weight *W* refers to the whole of the weight forward of the nodal point that revolves with the crankshaft, but weight at a considerable distance from the axis of rotation is of much greater importance than that closer to it, for the distribution of the weight determines the value of the least radius of gyration, *R*. Therefore, if the critical speed is to be raised it is highly important that the weight represented by the connecting rod head, the crank pin and the outer ends of the crank arms be kept down.

It has so far been assumed that it is desirable to so design the crankshaft assembly that the critical speed will be comparatively high. This assumption has a justification in the fact that any critical speed outside the operating range cannot cause any trouble. If a shaft for a six or eight-cylinder engine were made very light, while large valves and light reciprocating parts were used, so that the engine could run at high speeds, the engine might possibly be run up to the first critical speed, at which the crankshaft oscillates through one complete cycle during the time between explosions. This would be exceedingly troublesome, and shaft breakage would be likely unless special damping means were provided. There is, however, no difficulty in keeping the first critical speed beyond the operating speed range.

The second critical speed, which is exactly one-half the first, is within the working range in all modern six and eight-cylinder engines, it seems. It was stated previously that the tests made by Summers, which cover a great many different makes, show it to vary between 1800 and 2700 r. p. m. Engines whose second critical speed is near the upper limit of the range usually are capable of running at well beyond 2700 r. p. m. As long as the critical

speed is within the working range it does not seem to make very much difference whether it is near the limit or well within it.

An instrument for recording torsional vibrations has been developed by Mr. Summers and is illustrated in Fig. 7. It consists essentially of a short shaft carrying a disk, this assembly being attached to the forward end of the crankshaft (or an extension of it) by means of a collet. A flywheel driven by soft springs floats on this shaft so it rotates at the average speed of the shaft, while the disk, being rigidly connected with the shaft, follows any torsional vibrations of the crankshaft. A lever arm, one end of which carries a brass needle, the other end being forked, is connected by means of pivots to the disk and the small flywheel, respectively. Any difference in rotational velocity of the disk and flywheel causes the brass needle to move in a nearly radial line. This motion, combined with the rotation of the entire assembly, causes the brass needle to trace a figure as shown in the diagram. The brass needle has at its base a small disk painted white, which shows up against the black background, so the operator readily observes the form of the curve which the needle is tracing. This enables him to locate the critical periods and to determine the speeds at which he wishes to take diagrams for permanent records.

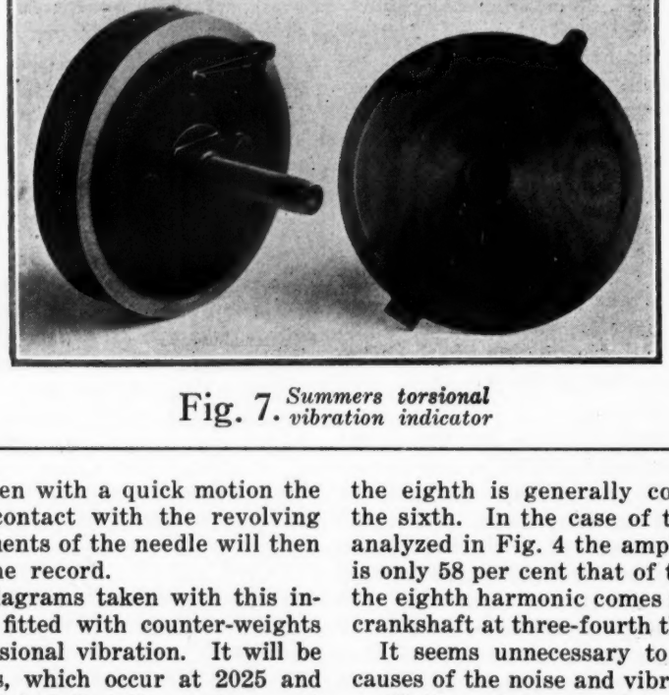


Fig. 7. *Summers torsional vibration indicator*

A disk of paper, treated on one side so contact with a brass needle will leave a trace, is held in a record-holder, consisting of a centering means, a disk and a hinged member for securing the record to the holder. This device guides on a forward extension of the indicator, and the operator holds it in readiness until the speed and other conditions are such as desired, when with a quick motion the paper disk is pushed into contact with the revolving needle. A trace of the movements of the needle will then appear as a black line of the record.

Fig. 8 shows a series of diagrams taken with this instrument from a crankshaft fitted with counter-weights but no means of damping torsional vibration. It will be noted that the critical speeds, which occur at 2025 and 1350 r. p. m., are well defined, but that the engine speed does not have to differ much from these critical speeds to reduce the vibrations materially. The product of the revolutions per minute and the number of cycles per revolution is the same for all of the critical speeds of the shaft, namely, 12,150. This is the number of complete cycles through which the crankshaft passes per minute, which corresponds to 202.5 complete cycles per second. It was found that by removing the counter-weights the frequency of vibration of the shaft was raised to 225 cycles per second.

Another instrument for tracing records of torsional vibration has been devised by Dr. Geiger, of the German M. A. N. firm. This instrument traces a coordinate diagram in which the ordinates represent the amplitude of the vibrations and the abscissas the crankshaft speed. Part of a "torsiogram" taken with a Geiger "torsiograph" from a Diesel engine is shown in Fig. 9.

The orders of the harmonics indicated are those of the gas pressure torque, and since the sixth and ninth harmonics produce unusually severe vibrations, it is to be inferred that the engine was a six-cylinder one. It has been observed that harmonic vibration is particularly pronounced in multi-cylinder Diesel engines, which is probably due to the high gas pressures of these engines.

Effect of Cylinder Number

As regards the effect of cylinder number, it has been claimed by European writers that torsional vibration is equally intense in four-cylinder as in six and eight-cylinder engines, only it is more noticeable with these latter because at points between critical speeds they run so much more smoothly than a four-cylinder. Not much trouble from torsional vibration in four-cylinder engines has been reported in this country, but this may be explained by the fact that we have built very few really high speed four-cylinder engines.

One would naturally expect the eight-cylinder to give more trouble from torsional vibration than the six, because of the greater length of its crankshaft, and in general the torsional rigidity of a crankshaft for a vertical eight is undoubtedly less than that of a six of the same displacement and designed along generally similar lines. There is a compensating feature in favor of the eighth, however, and that is that whereas with the six-cylinder the sixth harmonic of the gas pressure torque is the lowest for which the harmonics on all of the cranks are in phase, and, therefore, add together, in the case of the eight-cylinder it is only the eighth harmonic and the amplitude of the eighth is generally considerably less than that of the sixth. In the case of the gas pressure torque curve analyzed in Fig. 4 the amplitude of the eighth harmonic is only 58 per cent that of the sixth. On the other hand, the eighth harmonic comes into synchronism with a given crankshaft at three-fourth the speed as the sixth harmonic.

It seems unnecessary to go at length into the direct causes of the noise and vibration that accompany torsional oscillation of the crankshaft. The uniform (along the length) rotary motion of a six and an eight-cylinder crank train is completely balanced so far as the primary and secondary inertia forces are concerned, but super-imposed upon this uniform motion we have an oscillating motion which increases in amplitude from one end of the shaft to the other. This oscillating motion naturally produces reactions on the engine frame and hence sets it into vibration.

Of even greater importance than the speeds at which synchronous vibrations occur is the amplitude which may be assumed by them, as the only effective way of dealing with these vibrations consists in reducing their amplitude in some way.

The amplitude depends upon the exciting force. As the amplitude increases the energy consumed per cycle in overcoming the damping force increases, and when an exciting force is applied the amplitude will increase until

the amount of energy thus dissipated is equal to the energy supplied per cycle by the exciting force. Thus the amplitude of the vibration can be reduced either by reducing the exciting force or by increasing the damping force.

As regards the exciting forces, it is obvious that the various harmonics increase in amplitude with the area of the indicator card or the mean effective pressure. Of course, not all of the harmonics would necessarily increase in direct proportion with the increase in mean effective pressure, but in a general way there is proportionality. Therefore, by restricting the gas flow at high speeds and by keeping down the compression ratio and thus limiting the m. e. p., trouble from torsional vibration can be checked. This is in agreement with the observations of engine manufacturers that trouble from "roughness" at critical speeds increases in proportion as an engine is "forced." However, keeping down the output from an engine of a given size does not appeal to the majority of the makers, and is contrary to the general trend in the industry.

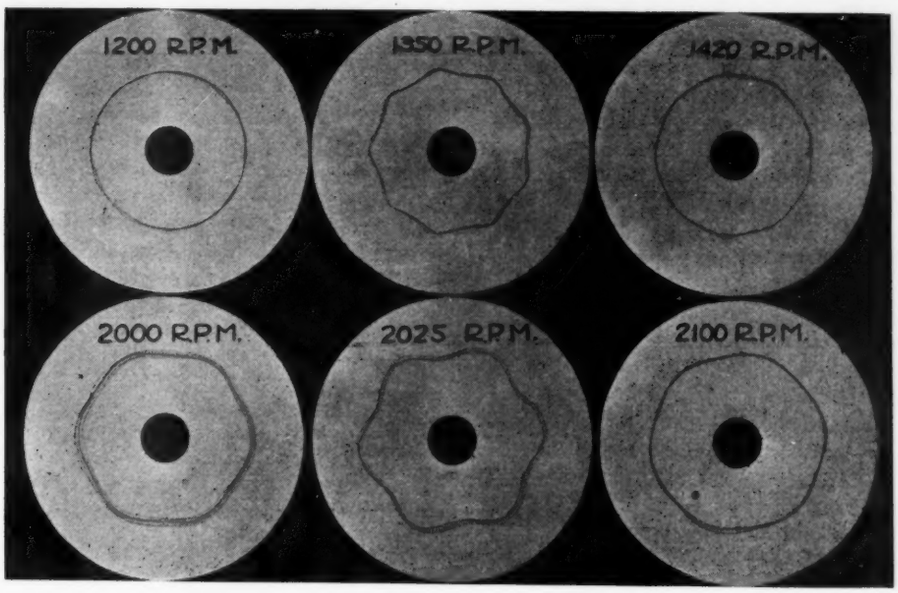


Fig. 8. *Torsional vibration diagrams traced by Summers' indicator*

The other method consists in increasing the damping force. It may be profitable to consider for a moment the nature of the damping force. If a crankshaft were performing torsional oscillations without rotating, then the bearing friction would constitute a damping force. However, since the oscillating motion is super-imposed upon a rotary motion and the rotary speed is at all times greater than the instantaneous value of the oscillating speed, the oscillating motion does not produce any additional relative motion between shaft and bearing. All that it does is to alternately accelerate and retard the shaft, and the motion of any point in the outer surface of a shaft of diameter d, relative to the bearing surface, is πd per revolution of the shaft, whether the motion is uniform or whether there is torsional vibration. If the bearing friction were a damping force, then tightening of the bearings, and consequent increase in bearing friction, should dampen out the vibrations, but it is the general observation that if the bearings are tight the engine is "rough," which is probably due to the fact that the shaft and bearings are never perfectly cylindrical, so that when the bearings are tight they are not uniformly tight for all positions of the shaft, and the varying friction will act as a vibration = exciting force.

There is undoubtedly a certain amount of molecular friction when the shaft is twisted, which acts as a damping force. That such friction exists when the shaft is stressed beyond its elastic limit is well known and generally admitted. But the general view in the past has been that when steels and other elastic materials are stressed within their elastic limit, they are perfectly elastic in behavior; that is, when the stress is removed the material returns to its original state and in so doing gives back all of the energy stored in it while being stressed. However, recent researches tend to discredit this view. There is, in fact, a certain hysteresis effect for all intensities of stress.

Character of Damping Forces

The term hysteresis is most readily explained by reference to magnetic phenomena. If we apply a certain magnetizing force to a piece of steel we produce in the steel a certain intensity of magnetic flux. If we then remove the magnetizing force the field intensity or magnetic flux decreases greatly, but it does not drop to zero; in other words, the steel does not return to its original magnetic state. To bring it back to this state we must apply a reverse magnetising force of a given strength, which means

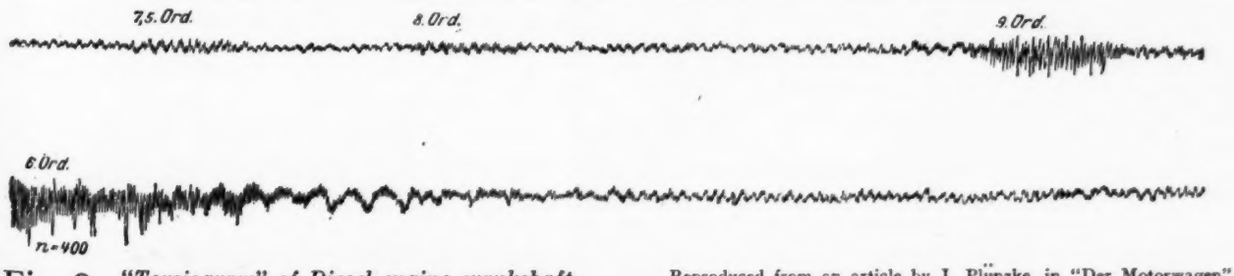


Fig. 9. *"Torsiogram" of Diesel engine crankshaft traced by Geiger "torsiograph"*

Reproduced from an article by J. Plünzke, in "Der Motorwagen"

that we must expend energy. If a piece of steel is subjected to an alternating magnetizing force we must expend this amount of energy twice per cycle, and the energy loss or hysteresis loss is represented by the area of the hysteresis diagram shown in Fig. 10.

Similarly, if we subject an elastic body to cyclical stresses, the strain or deformation corresponding to a certain stress will not be the same when the stress is decreasing as when it is increasing. When the stress is removed, a certain slight deformation remains and energy must be spent to eliminate it. Thus if stress and strain or stress and deformation were plotted against each other we would obtain a diagram very similar in shape to a magnetic hysteresis diagram, and the area of this diagram is a measure of the energy loss. This loss assumes measurable proportions only when the stress reaches the elastic limit, but there is a slight hysteresis loss even for very small stresses, hence molecular friction enters into play as a damping force.

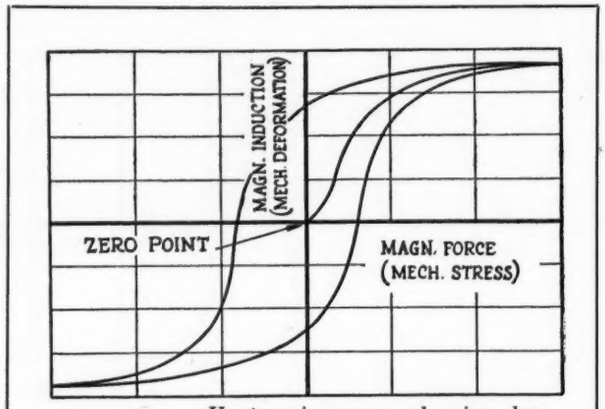


Fig. 10. *Hysteresis curve showing loss due to alternating forces*

There is also a slight increase in the energy loss due to the churning of the air by the crank and its attached parts. Let us assume that a crankshaft passes through speed cycles of a minimum value v—a and a maximum value $v + a$. Then the average speed is v, but the air resistance is proportional to the square of the instantaneous speed, and the mean value of the squares of the instantaneous speeds is not v^2 but $(v^2 + a^2/2)$. For instance, if the speed varied between 50 and 150 per cent of the mean, the friction would increase in the proportion of

$$(1^2 + 0.5^2/2) - 1 = 0.125$$

or 12.5 per cent. The power loss due to the friction would increase in even greater proportion, and air friction as well as any other disturbances of a turbulent nature, therefore, will act as a damping force.

These forces naturally are very small and, therefore, at critical speeds the vibrations are likely to reach excessive amplitudes if no artificial damping forces are applied. Any device which absorbs energy when there is torsional vibration or irregular motion, more or less in proportion to the amplitude of the vibration or the irregularities, will serve to produce this damping force. The most widely used device is the Lanchester vibration damper which consists merely of a small flywheel carried at the forward end of the crankshaft and connected thereto by a friction clutch or coupling which slips (and absorbs energy) whenever the forward end of the crankshaft accelerates or decelerates appreciably.

There are really three detrimental effects of torsional vibration. The first is the "roughness" or noise of the engine which has been repeatedly referred to and which is annoying to the passengers, while at the same time it is indicative of stresses which undoubtedly accelerate the wear of the engine in general. The second is the great stress imposed on the so-called front end drive by the irregular motion. In cases where there is a very pronounced tendency toward torsional vibration it is quite impossible to use for the front end drive the more rigid types of transmission, which naturally are more sensitive to irregularities in speed. The third effect is irregularity in the drive to the gearbox and the rear axle.

In certain discussions it is assumed, for the sake of simplicity, that the engine flywheel turns at uniform angular speed. However, only a flywheel of infinite moment of inertia would run at uniform speed when subject to a variable crank moment, and in all practical engines there are slight fluctuations in flywheel speed. These fluctuations are materially increased, however, when the engine runs at a critical speed. It was shown previously that the nodal point is located some short distance ahead of the flywheel and that the flywheel is subject to torsional vibration the same as the forward end of the crankshaft. As long as the nodal point is very close to the flywheel the amplitude of the latter's vibration will be small, and its effects will be negligible. If the nodal point is moved forward or further away from the flywheel, this amplitude is increased. This can be done by adding weight to the crank throws or at the forward end of the crankshaft, or by reducing the weight (or, rather, the moment of inertia) of the flywheel. If a light flywheel is used the irregularity of the motion of the driving end of the crankshaft is often sufficient to cause "gear-clatter" in the transmission. Many manufacturers have experienced this trouble, which can be remedied by using either more effective flywheels or incorporating a flexible member, such as a rubber cushion or a fabric driving disk, in the drive.

It is an unfortunate thing that the fitting of balance weights—designed to reduce bearing loads—lowers the natural period of torsional vibration of the crankshaft, That it should do this is directly apparent from the equation for the natural frequency of vibration given in the foregoing. The balance weights add to the value of W and as the frequency is proportional to the square root of the reciprocal of W, if the balance weights were made large enough to completely balance the reciprocating and rotating masses, the frequency of the torsional vibration of the shaft would be decreased almost in the ratio of

$$\sqrt{\frac{1}{1}} \text{ to } \sqrt{\frac{1}{2}} = 1{:}0.707.$$

Such large balance weights are rarely used, however. The diagrams shown in Fig. 8 are from a crankshaft assembly with balance weights, and the frequency of this crankshaft assembly was increased from 202.5 to 225 cycles per second by removing the balance weights. This corresponds to a reduction in the frequency—that is, of the critical speed—in the proportion of 1 to 0.9 by the balance weights.

In the foregoing, reference has been made repeatedly to the amount of energy supplied to a torsionally vibrating shaft per cycle, and it was shown that the maximum amplitude of vibration at critical speeds is proportional to this amount of energy. If an exciting moment of amplitude T is applied to the forward end of the crankshaft, the resulting flexure of the shaft will be

$$\theta = T\,l\,/\,E\,J \text{ radians.}$$

where l is the length of the crankshaft forward of the nodal point; E, the torsional modulus of elasticity, and

J, the polar moment of inertia of the shaft. The energy stored in a shaft by torsional deflection is equal to $T\theta/2$, hence

$$\text{Energy stored} = \begin{matrix} T^2 l \,/\, 2\,EJ \text{ inch-pounds or} \\ T^2 l \,/\, 24\,EJ \text{ foot-pounds.} \end{matrix}$$

Thus the chief factor determining the amplitude of vibration is the amplitude of the particular harmonic of the gas pressure turning moment which happens to be in synchronism with the shaft considered as a torsional pendulum. In fact, the amplitude of the vibration varies as the square of the amplitude of the exciting harmonic, and inversely as the rigidity of the shaft. The torque harmonics cannot be reduced without decreasing the power, and aside from the application of damping forces the only means of reducing the amplitude of the vibrations consists in reducing the length of the shaft and increasing its moment of inertia, that is, its diameter.

The mass of the crank arms, etc., has no effect on the amplitude which will be reached by the torsional vibration, except insofar as it may affect the damping force. The damping force inherent in the shaft is a factor that is very difficult, if not impossible, to express in mathematical terms.

Control of Vibration

Of the various methods of controlling torsional vibration, that by the use of the Lanchester vibration damper is most familiar. This consists of a small flywheel carried at the forward end of the crankshaft and held thereto by frictional members. At each impulse or change in torque, there is a slight slippage at the frictional surface, and the energy thus consumed is abstracted from the crankshaft, so that it cannot build up its torsional vibration to anything like the same amplitude as would otherwise be reached.

Some manufacturers use a small flywheel at the forward end, in addition to the regular flywheel at the rear. The addition of a flywheel at the forward end—everything else remaining the same—has the effect of moving the nodal point forward. The critical speeds are lowered somewhat because the pendulum weight is added to without the torsional stiffness being changed. On the other hand, the amplitude of vibration at the forward end may be reduced, because the factor l, that is, the distance from the nodal point to the forward end of the shaft, will be reduced. To make this clear it is only necessary to assume that the flywheel at the forward end be of great moment of inertia as compared with all the other revolving masses, in which case the forward end would rotate at nearly uniform speed. Thus it seems reasonable to expect smoother operation of the front end gears when an extra flywheel is used at the front.

On the other hand, moving the nodal point forward involves greater flexibility of the rear end of the shaft, and therefore greater irregularities in the speed of the flywheel. If these irregularities become excessive there is likely to be "gear clatter" in the transmission. That is, every time the flywheel accelerates and decelerates the slight clearance between the teeth of the transmission gears will be taken up. The irregularities in the rotary speed of the flywheel depend upon the moment of inertia of the rear flywheel, and when a front flywheel is to be used it would be a mistake to reduce the moment of inertia of the rear flywheel correspondingly. This gear clatter referred to is a phenomenon well known in the industry, occurring when unusually light rear flywheels are used in conjunction with a crankshaft of considerable inertia (due to a long stroke, counterweights or a front flywheel). The usual remedy is to use a flexible member in the transmission line, generally between the clutch and gearset, which absorbs the irregularities of motion and prevents the breaking of contact at the gear tooth surfaces.

From some points of view a flywheel at the center of the engine is a desirable arrangement. This places the nodal point at the center, reduces the length of the free swinging portion of the shaft by about one-half, and the weight of the swinging masses correspondingly. Hence, the natural rate of vibration will be much higher and the amplitude smaller. However, it is rather difficult with the present type of block-cast engine to accommodate a flywheel of adequate size at the center.

Harmonic Balancer

A device of comparatively recent development designed to control torsional vibration is known as the harmonic balancer. It consists of a swinging balance weight located alongside of a long crank arm and pivoted to it. The balancer is held in a central position relative to the crank arm by a leaf spring and can swing through a small angle around its fulcrum. The harmonic balancer is being used on the present Oakland model and the following explanation of its action is given by C. E. Summers of the General Motors Research Laboratory, who originated the device:

The harmonic balancer operates on the principle of interfering vibrations. It controls torsional vibration of the crankshaft by opposing the oscillatory movements of the shaft by movements of another body. This body is in the nature of a torsion pendulum attached to, and having a frequency approximately the same as, the crankshaft assembly. When the piston impulses come into resonance with the shaft the torsion pendulum vibrates out of phase with the crankshaft and thus opposes the vibrations of the latter.

A modification of the Lanchester vibration damper has been patented recently by the Austro-Daimler Co. (British patent 217,928, issued Sept. 23, 1925). To the rear end of the crankshaft is keyed a clutch drum or housing which has a coned outer surface upon which is mounted the flywheel. The latter has a hub bored to a corresponding taper that is pressed against the conical surface of the clutch housing by a spring. The object of this frictional mounting of the flywheel is to damp out any vibrations of the driven shaft.

Also a Problem in Ship Design

It is interesting in this connection to point out that automobile engineers were not the first to have to deal with torsional vibration. The same phenomenon occurs in steamships and motorships, where the engine at one end and the propeller at the other of a long line of shafting form a torsional pendulum of comparatively low natural period of oscillation. The speed of revolution of large marine engines also, is low, and in extreme cases of torsional vibration the connecting rods of these engines appear to stand still for an instant near the middle of the stroke. Marine engineers have gone into the subject at great length, and many methods have been developed for pre-determining the natural rate of oscillation of a line of shafting with engine and propeller. Where such large units are involved it is, of course, far more difficult than in automobile work to go into the subject experimentally, and it was, therefore, highly important that methods be developed for determining the critical speeds in advance. Moreover, the torsional rigidity of a long line of plain shafting can be calculated with much greater accuracy than that of a multi-throw crankshaft.

Chief Lacquer Issues Are *Colors, Luster* and *Methods of Application*

Opinion varies greatly among lacquer makers and users as to how the present finishes are meeting requirements and how they should be applied.

By K. W. Stillman

ABOUT the only thing which is certain in respect to the present situation of automotive lacquers appears to be the uniformity of disagreement among lacquer makers and users as to how present lacquers are meeting the requirements and how they should be applied.

As a result of an extensive survey of both makers and users of lacquers made by *Automotive Industries*, the following items appear to be accepted as true by a majority of those questioned, although on none of them is there a unanimity of opinion.

1. In general, enough colors are now available. Tones and shades of red are inclined toward muddiness; blues are not durable; maroons are unsatisfactory.
2. Mixed colors are more troublesome than single pigment colors.
3. Inorganic pigments are best for lacquers and new pigments may be developed especially for lacquer work.
4. Improved methods of making and grinding pigments and of applying finishes may eliminate many of the present difficulties.
5. Varnish top coats add luster, but at the expense of many of the advantages of lacquer finishes.
6. Clear lacquer top coats are impracticable with present methods of application. Durable clear lacquers are possible for the future.
7. High gloss finish is not in great demand; well rubbed lacquer is satisfactory to the public.
8. Lower air pressures in spray guns would eliminate much of present necessary sanding and polishing work.
9. Satisfactory pyroxylin primers are available but there is a question as to their relative advantages over oil primers.
10. Flow coating or brush applied top coats are future possibilities for obtaining luster with less effort.

As stated before, although these items appear to be accepted by a majority of the persons questioned, there are others who disagree with the statements and can present strong arguments for their side of the case. As impartially as possible this article will present the arguments offered on both sides of the questions, since it still appears to be too early for anybody to arrive at a definite decision regarding the value of the many claims made.

In refutation of the statement that some colors are not satisfactory it is pointed out that more colors are being used by car makers than ever before in the history of the industry. There is scarcely a maker who does not have at least one model finished in blue so that some of the available blue pigments, at least, must be satisfactory. The number of entirely satisfactory red pigments is generally conceded to be limited but some makers contend that there are plenty to supply all reasonable demands of car manufacturers. Maroons have been found to be unsatisfactory

by most lacquer and car makers but at least two well known lacquer companies claim to have developed very durable finishes of this color.

Possibly the main objection to present blues is caused by the fact that ultramarine, which is generally considered superior in appearance to other blues, is not suitable for lacquer work and Prussian or Chinese blues have been used instead. These pigments, while comparing favorably with most others, are usually considered below ultramarine in appearance. The darker shades of blues appear to give the most trouble, while light reds are the most prevalent cause of dissatisfaction in that color field.

Although the opinion is almost unanimous that mixed colors cause more trouble than single pigment colors, many of the lacquer makers say that this is not the fault of the pigments or of the lacquers but is due to the methods of using them. It frequently happens that the pigments used vary considerably in specific gravity so that unless the lacquer is violently agitated before and during use the heavier pigments tend to settle to the bottom and the finish will be unsatisfactory in color and in structure.

To overcome this difficulty some lacquer makers see the future possibility of the development of organic pigments which will be of approximately equivalent specific gravity. Other makers, however, see no need for such a development but insist that users of lacquer must learn better methods of application.

Properly mixed colors should, in general, give as complete satisfaction as those obtained by the use of a single pigment and while getting absolute color matches between batches will be more difficult with mixed colors it should be no more so than was true with paint products. Practically all lacquer makers unite in saying that troubles of this kind experienced by lacquer users are almost always caused by application methods.

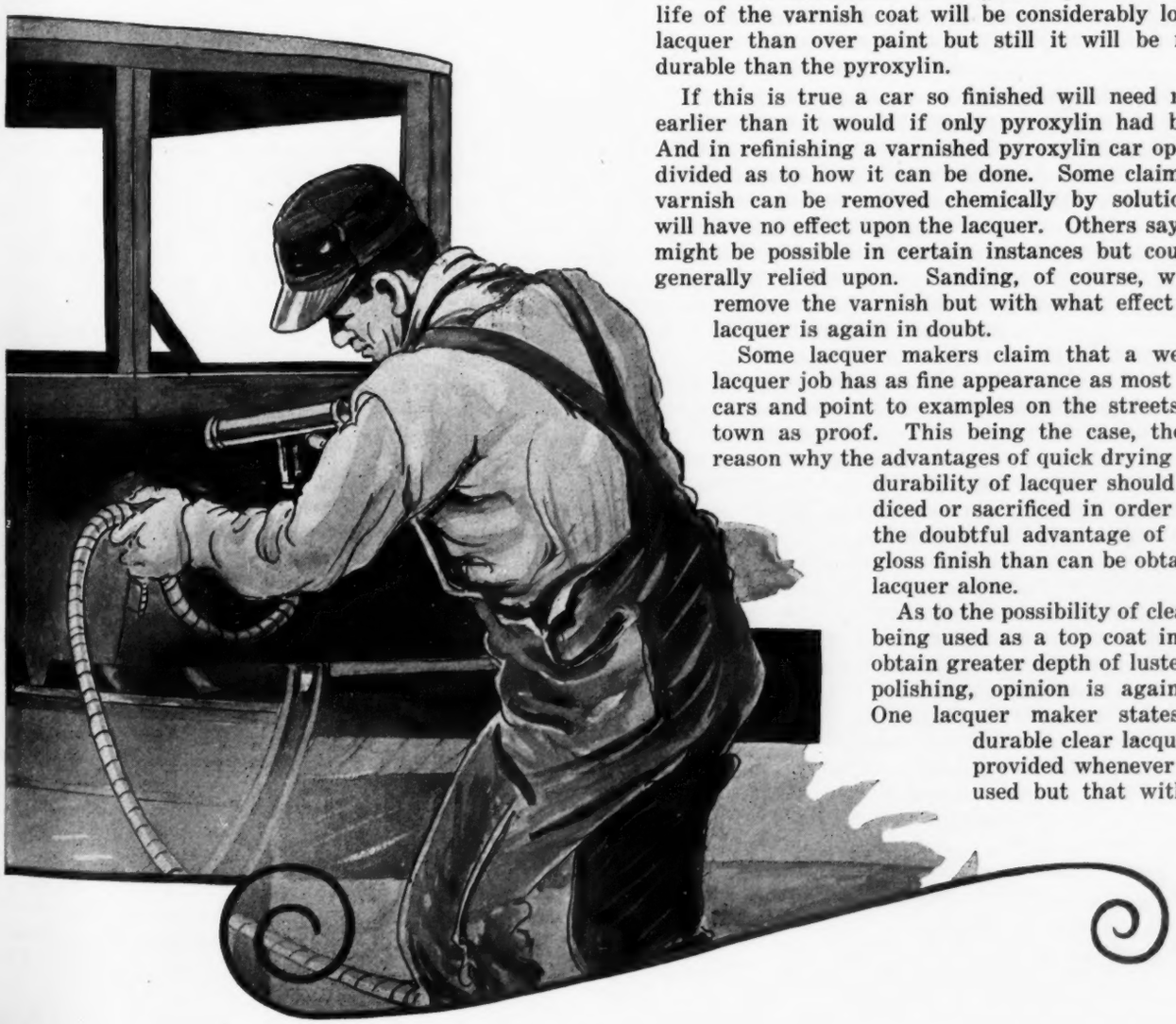
Present lacquers and pigments are considered to be basically good but detail refinements are possible in their manufacture and great improvements are highly desirable in application methods.

Complain of Blues Chalking

Several users complain of blues chalking off. One lacquer maker states that this tendency is no more pronounced in blues than in other colors but that the high tinctorial strength of blues makes their staining action more evident on the polishing cloth and thus lead the uninformed to the conclusion that his finish is deteriorating rapidly. This maker says that after a series of tests with pigments which have this chalking tendency much more than any blues on the market the conclusion has been reached that this surface erosion of the pigment can be removed by polishing and still leave a good finish after several years of ordinary use.

Although at least one well known lacquer maker recommends the use of a varnish top coat for obtaining high gloss finish and at least one prominent car manufacturer is applying a varnish coat over pyroxylin under coats with apparent success, the prevailing opinion is that such practice is not advisable. It is generally conceded that the life of the varnish coat will be considerably longer over lacquer than over paint but still it will be much less durable than the pyroxylin.

If this is true a car so finished will need refinishing earlier than it would if only pyroxylin had been used. And in refinishing a varnished pyroxylin car opinions are divided as to how it can be done. Some claim that the varnish can be removed chemically by solutions which will have no effect upon the lacquer. Others say that this might be possible in certain instances but could not be generally relied upon. Sanding, of course, will always remove the varnish but with what effect upon the lacquer is again in doubt.



Some lacquer makers claim that a well rubbed lacquer job has as fine appearance as most varnished cars and point to examples on the streets of every town as proof. This being the case, they see no reason why the advantages of quick drying and great durability of lacquer should be prejudiced or sacrificed in order to obtain the doubtful advantage of a greater gloss finish than can be obtained with lacquer alone.

As to the possibility of clear lacquer being used as a top coat in order to obtain greater depth of luster without polishing, opinion is again divided. One lacquer maker states that a durable clear lacquer will be provided whenever it can be used but that with present

methods of application there would be no advantage in it since the force with which the lacquer film is deposited would still cause an orange peel effect and would require polishing.

One or two other makers claim that durable clear lacquers are now available and have been used with satisfaction. Still others say that such a development is highly improbable because of the susceptability of pyroxylin to the destructive effects of the sun's rays. An attempt to protect the cellulose by use of soluble dyes would fail, they say, because of the dyes' lack of durability.

Still another school of thought on this subject is not greatly concerned with whether such a development is possible or not but confines its comments to statements that it is not necessary—that present lacquers are entirely satisfactory and that as application methods are improved the labor of obtaining high gloss will decrease. These companies offer the suggestion that lacquer users may be demanding too much—that they should be more satisfied with what they have since, in view of the widespread adoption of lacquer finishes, it must be meeting all demands better than any finish ever has before.

Greater Care in Application

It is generally believed that, should a durable clear lacquer be developed, much greater care would be necessary in applying it than is now given to lacquer application in order to obtain all the beneficial results possible from its use.

That high gloss finishes are declining in popularity is the opinion of both car and lacquer makers, although there are some members of each group who believe that, given a choice of two cars otherwise the same, the average prospect would choose the one with the higher gloss.

A well rubbed lacquer finish is claimed to be the equal in luster of the average paint and varnish job by a number of companies. Of course, while this point may be conceded by car makers, they usually feel that the labor cost of obtaining such luster through an extensive polishing operation is excessive, or at least is large enough to warrant considerable research in developing methods or materials to reduce it.

One suggestion which offers some possibilities has to do with the durability of lacquer finishes. This factor was the important one when lacquers were first used in the automotive industry and nearly all development work was aimed toward increasing the life of the finish. Some of the finishes now in use may possess a longer life than is necessary—that is they are likely to wear longer than the car itself or, at least, to outlast the desire of the owner to keep a particular color combination.

If this proves to be true, an opportunity to increase the ease of polishing is offered by permitting larger quantities of gums and resins to be used in the finishes. Increasing the gum and resin content makes polishing easier but lessens durability. When more experience in regard to the life of lacquers as now constituted is obtained it may be found practicable and desirable to decrease their life to some extent in order to increase their luster.

Development of flow coating is another possibility suggested by several lacquer makers. At least one maker claims to have a lacquer now which is suitable for this work, but again development is hindered by lack of proper methods of application.

Brush application of the final coats is another suggestion made by one lacquer maker which seemed to find little support but which might be developed into something of value.

One item upon which all lacquer makers appear to agree is that sanding and polishing can never be eliminated on lacquer surfaces. Such work was always necessary in order to obtain a good paint and varnish job and it is much more necessary when working with lacquer. While a paint and varnish finish has a tendency to cover up defects and unevenness in the surface upon which it is applied, the nature of a pyroxylin film has just the opposite effect and all such defects are greatly magnified by a pyroxylin coating. Since it is unlikely that steel makers will ever be able to produce an absolutely smooth steel and, even if they could, press marks and particles of dust would still be present on the sheets of metal, it is improbable that a satisfactory lacquer finish will ever be possible without a certain amount of polishing.

Although some car makers are obtaining excellent results with relatively high air pressures it is the general belief of lacquer makers that the use of lower pressures in spray guns, if the latter were designed to give proper atomization, would produce better results and that the increased time consumed in applying the finish would be more than offset by decreased time of polishing made possible by the smoother coats.

A large majority of lacquer makers believe that oil primers are much better than pyroxylin and, in fact, some of them doubt the possibility of using a lacquer primer satisfactorily.

The objections raised to lacquer primers are numerous and nearly every opponent has a different one. One asserts that oil primers should be used because they have a greater affinity for the metal than for the lacquer surfacers so that in case of an accident the finish would split between the two coats and leave the primer to protect the metal from moisture.

Another states that the solvents in lacquer primers become trapped in surface cracks and particularly around moldings and such places and when they finally escape cause blisters on the surface of the finish.

Several lacquer makers, however, are very much in favor of lacquer primers and they have been used by a number of car makers with apparent satisfaction. The chief advantages to be gained are cited by this group as being the great saving in time, labor and fuel possible with the use of a complete air-drying nitro-cellulose system. Advocates of lacquer primers say that such materials, when properly made, adhere to metal quite as well as the best of oil primers, that they give no trouble through shrinkage, and that they can be given the same relative coefficient of elasticity as the rest of the lacquer film so that there will be no difficulties from this source.

Advantages and Disadvantages

A summary of the relative advantages and disadvantages was made by a representative of a company which has both a lacquer primer and an oil primer developed for use with pyroxylin finishes. He said that it is possible to obtain a pyroxylin primer which is as durable as an oil primer and which will adhere to the metal as well if not better. It also can be applied much quicker and needs no oven drying. To offset this advantage of time saved are the items of increased cost of materials and the much greater care which must be given to cleaning the metal.

For installations where there is oven equipment available this company recommends the oil primer but in cases where there is no oven equipment they believe that saving the cost of ovens will usually offset the disadvantages of lacquer and advise its use as a primer. In other words, both lacquer and oil primers are quite satisfactory as a basis for a lacquer finish. Which to use depends more upon the surrounding conditions influencing its application than upon any other factor.

Bus Engineering Problems Complicated by *Restrictive* State Laws

Analysis is made of legislative and regulatory limitations which manufacturers must take into consideration in design and equipment of motor buses.

BECAUSE of the very rigid and detailed restrictions which have been placed upon bus design and construction during the past few years by State legislatures and regulatory agencies, some rather difficult engineering problems must be solved by the manufacturers in this field if their vehicles are to be saleable in several States without undergoing major alterations.

This situation makes it necessary that the bus manufacturer be informed of what limitations have been placed upon his work in each State. On the next page is a table which gives this information in readily available form.

Inasmuch as there appears to be considerable difference in the relative permanency of restriction authorized under acts of the legislature and those imposed by the regulatory commissions they have been differentiated in the table. Items in Roman type are restrictions imposed by State legislatures and, of course, can not be repealed or altered except by passage of another bill. Items in *italics* are restrictions imposed by State commissions and these rules, in general, can be repealed or altered at the will of the commissions and without notice.

How Information Was Obtained

The tabulation presented here is as nearly complete and accurate as it is possible to make it. Copies of rules and regulations have been obtained from both State Motor Vehicle Departments and from the commissions regulating public utilities, where such exist. In a number of States the legislatures are meeting this year and the laws regarding motor vehicles may be changed and, of course, the public utility commissions can add new regulations or alter old ones at will. With these possible exceptions the restrictions reported here may be considered complete and accurate.

Speed restrictions vary from 20 miles per hour to 45. In nearly every State the law reads that the speed shall not be great enough to endanger life or property and then adds that a speed of more than 35 miles per hour—for example—will be considered *prima facie* evidence that the law is being violated. In the absence of specific restrictions upon the speed of buses the allowable speed as given in the above connection has been used as the maximum in the table.

There is considerable variation in the methods used for determining license fees and consideration given to this subject when designing a bus may permit considerable economies to be gained in operating expenses. Some States do not differentiate buses from trucks or passenger cars in determining license fees. In States where buses are under special regulation, seating capacity or mileage, or both, are usually employed, often in connection with weight, in setting the fee. The sliding scales used in many States should be very carefully studied, however, so that a slight addition in weight or seating capacity does not increase the fee out of all proportion to the increased weight advantage.

The Law in Maryland

Possibly the worst—or best—example of what might happen in this connection is to be found in Maryland where license fees are based upon seating capacity, route mileage and vehicle weight. The scale of fees is so adjusted that while a 25-passenger bus operating 40,000 miles per year would pay a license fee of $550 if it weighed 8000 lb., if its weight were increased to, say, 10,000 lb., its tax under identical operating conditions would be increased nearly three times to over $1400.

Standardization of bus designs, to some extent at least, is desirable from every viewpoint but the present status of regulations quite effectually prevents any such movement being carried out. As the situation now stands conflicting regulations imposed by the several States make it necessary to build a special body, in some respects at least, for sale in each State. That this is highly uneconomical, not only to manufacturers but to operators as well, is evident, but such methods will continue to be necessary so long as the various State regulatory bodies hold such widely divergent views as to what constitutes a properly built and equipped bus.

In the last column of the table will be found abundant evidence of the tremendous interest which State authorities have come to have in bus operation. That much of this interest has not been shown in ways conducive to better conditions in the manufacturing industry is unfortunate and is a compelling reason for the bus industry to take a strong stand for a higher degree of uniformity in the restrictions than now exists.

Regulation of bus operation is probably advisable and the bus industry has no serious objection to it in theory. It has become obvious, however, that the regulatory bodies are going too far in some cases.

Legislative and Regulatory Restrictions Upon Bus Design and Construction

Note: Items in Roman are State Laws. *Items in Italic are Commission Rulings*

STATE	Maximum Length in feet	Maximum Width in inches	Maximum Height in feet	Maximum Weight (Including Load)—in pounds	Max. Weight per Axle. (Including Load)—in pounds	Max. Weight per inch width of tire—in pounds	Max. Weight per wheel—in pounds	Maximum Speed Miles per hour	Is a Mirror Required?	Color of Tail Light	Max. Candle Power of Headlights	Basis of Registration Fees	Are Emergency Doors Required?	What are the interior lighting requirements? Minimum c. p. of lamps	Is a Speedometer required?	Is an Extra Tire required?	Is a Fire Extinguisher required?	Is a Destination Sign required?	What other requirements are there?
Alabama				20,000				30		Red		Seat Cap.							
Arizona	30	96	12½	28,000	22,400	800		25		Red	32	Horse power							Direction Indicator.
Arkansas	30	96	12½	22,000-2 axles 28,000-4 axles		350 to 765	700 to 10,720	35	Yes	Red									
California		96		22,000-4 wheels 34,000-6 wheels	18,000	700	9,000	35	Yes	Red				*Reqd.-2 c.p.*	*Yes*	*Yes*	*Yes*		
Colorado				16,000(load only)		700		35		Red		Seat Cap.							
Connecticut	40	96		25,000	20% gross load (Min.)	800		30		Red Min. 6 c. p.	21	Seat Cap. plus piston displacement or load capacity—whichever is greater.	*On left rear side*	*12*	*Yes*		*Yes*	*Yes*	*Exhaust whistle on interurbans; seating capacity plate; two purple lights with lenses not less than 3 in. diam., and lamps not less than 6 c.p. displayed with destination sign; light or buzzer annunciator from emergency door to driver; tail light lense not less than 3 in diam., lamp not less than 6 c.p.; yellow stop lights with lenses not less than 4 in. diam., lamps not less than 21 c.p. connected with both hand and foot brakes.*
Delaware		96	12½	22,000	16,000	700		35		Red		Total weight Load-cap. x 125 lb.							
Dist. of Col.	30	96	12½	28,000	22,400	500 to 800		22	Yes	Red	About 21		*Yes*	*Required*				*Yes*	*Ventilators; Heaters;* Lock (Wheel locks prohibited)
Florida		84	12	16,000		600		25				Gross weight and seat cap.							
Georgia								30		Red									
Idaho						500		30		Red									
Illinois		96		24,000 *125% rated cap.*	16,000	800		25	Yes	Red		Gross weight Load-Cap. x 125 lb.		*Reqd. 2 c.p.*		*Yes*			*Heating system.*
Indiana				28,000	22,400	800		30	Yes	Red		Capacity							Red Stop light.
Iowa		96		18,000 (pneu.) 14,000 (solid)		800	8000	*30*		Red	32	Load wt. (cap. x 150) plus mileage	*At rear or opposite side*	*Reqd.-2 c.p.*	*Yes*	*Yes*	*Yes*		*Windshield wiper; 2 red flags and lights.*
Kansas		96		Load 150% rated cap.	Cities have power to prescribe weight Limits			40		Red		Seat Cap.		*Reqd.-2 c.p.*		*Yes*			*Heating system.*
Kentucky		90				800		30				Cap. and weight							
Louisiana		84			15,000 (rear)	600		35	Yes	Red									
Maine		96	12½	18,000-4 wheels 25,000-6 wheels	13,500	700		35	Yes	Red	32								Capacity Platemarker.
Maryland		90		20,000		650		25	Yes	Red	32	Capacity, route mileage & weight						*On sides*	
Massachussetts	28	96		28,000		800		20	*Yes 2 outside and inside*				*Rear middle or side*	*Required to light interior, platform & steps.*	*Yes*		*Yes*	*Yes*	Yellow stop lights; *locks; front and rear marker lights at upper corners of body, red in rear, other color in front lenses not less than 2 in. diam.; yellow stop light; windshield cleaners; guard around driver; grab handles at doors and for standees; drive shaft protected to prevent it falling to ground or hitting bus floor if it becomes disconnected; ventilators; pneumatic or cushion tires; diagram of gear shift posted in view of driver.*
Michigan	40	96	12½	28,000	18,000	700		35	Yes	Red		Weight			Yes				

Michigan	40	96	12½	28,000	18,000	700		35	Yes	Red		Weight			Yes				3 green lights front and rear; mudguards 8 in. wide min.; 76 in. gage max.
Minnesota	30	96	12½	28,000	22,400	800		35	Yes	Red	32								
Mississippi				12,000 (Load only)				30		Red		Capacity plus mileage							
Missouri	30	96	12½	24,000-4 wheels 38,000-6 wheels	16,000	600		25		Red									
Montana										Red									
Nebraska	90	12				600	7,000	35		Red					Yes				
Nevada				25,000-2 axles 30,000-3 axles		600		45	Yes	Red	21	Vehicle weight and seat capacity							Speed limit 8 m. p. h. for venicles over 96 in. wide.
New Hampshire	30	96		20,000	15,000	750		25	Yes	Red		Weight and load (seat cap. x 150)	*In rear*						*Automatic windshield wiper.*
New Jersey	28 *24 (body) 16 min.*	96 *84 min.*	12½ *6½— 6½ inside clearance*	28,000		700	11,200	30	Yes—*2 outside and inside*	Red	21	Seat capacity and weight	*In rear center—24-in. clearance minimum*	*5 c. p. per seat passenger capacity minimum*			*Yes*	*Yes*	*Stop light; Window guards; guard rail beside driver; wood and glass partition behind driver; grab handles in vestibule; grab handles for standees; ventilators; heating system; fire-proof front foot boards; ¼ in. frame stock for 6 in. channel; ⅛ in. stock for 8 in. channel; 35 in. maximum frame height; pneumatic or cushion tires.*
New Mexico		96		12,000 (Load only)		700		25	Yes	Red		Weight and Seat cap.							
New York		96		28,000-4 wheels		700	11,200	30	Yes	Red	21		*At opposite end from service door*	*Required*			*Yes*	*Yes*	Side Capacity marker.
North Carolina		86		15,000				25		Red				*Required*	*Yes*	*Yes*			
North Dakota								30		*Green 3 c. p.*		Seating Cap.		*Required 2c.p. lamps-min.*	*Yes*	*Yes*	*Yes*		*3 c.p. green light at lower right side windshield; heaters.*
Ohio	30	96	12½	20,000	16,000	450 to 650		35	Yes	Red	32	Seating Cap. or vehicle wt.	*Side and rear or both sides*	*Required 2c.p. lamps-min.*	*Yes*	*Yes*			*Heating system;* 2 red flags and flash lights.
Oklahoma		90		15,000		800		35				Mileage		*Required for nightoperation*	*Yes*		*Yes*		
Oregon		96		22,000 (after Jan. 1, 1927—20,000)	17,600 (after Jan. 1, 1927—14,400)	500		30		Red		Vehicle weight and seating cap.			*Yes*		*Yes*		White light on left side of vehicles over 72 in. wide.
Pennsylvania	30	96	13	26,000	19,500	800		30	Yes	Red	21(reqd.)	Seating Cap.							
Rhode Island		102	12½	28,000	22,400	800		35		Red		Horse power and gross wt.			*Yes*			*Yes*	Lock; capacity plate.
South Carolina		90		20,000	15,000	600		35											
South Dakota				10,000 (Load only)				25											
Tennessee				20,000		650		30	Yes										
Texas										Red or yellow									
Utah		96		20,000		600	7,500	30		Red	32	Load Cap.							Direction Indicator; Load Plate
Vermont		96		16,000		600		25	Yes	Red		Weight and Cap. (Seat x 150)	Rear left or center				*Yes*	*Yes*	Lock.
Virginia	30	96	12½	20,000 (10,000-load only)		500 to 700		20						*Required 2c.p. lamps-min.*	*Ys*	*Yes*	*Yes*		
Washington	30	96		24,000	18,500	800		25	Yes	Red	27	Weight and capacity							Direction indicator; white light on left side of vehicles over 72 in. wide.
West Virginia	40	90	12	20,000 (load = seat cap. x 125)	17,000	600	9,000	35		Red	32	Capacity and mileage	In rear of pay-enter type					*Yes*	Load plate.
Wisconsin	30	96		24,000 (load = seat cap. x 150)	18,000	800		30	Yes			Weight and load							Load plate; double deck interurban prohibited.
Wyoming	30	96	12½	25,000	18,000	700		35	Yes	Red or green		Horse power and capacity							

Supercharging Investigated *from the* Thermodynamic Standpoint

An analytical study of the effects of supercharging on power output and brake thermal efficiency with discussion of general characteristics of supercharged engines for automobiles

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SUPERCHARGED engines can be defined as that type in which the charge of combustible mixture for each cylinder, i. e., the weight of fuel and of oxygen necessary for its complete combustion, which enters the cylinder during the intake period is increased beyond the normal quantity, that is to say, beyond the quantity corresponding to atmospheric pressure in the case of four-stroke engines, and beyond that corresponding to a slightly higher pressure (due to the air-pump) in the case of two-stroke engines.

The object of supercharging is to burn in each cylinder a quantity of fuel greater, and thus liberate an amount of heat greater, than would be normally possible, thereby increasing the power output of the engine without increasing the cylinder volume.

Historically, the idea of supplying the carburetor with air at above atmospheric pressure is more than 20 years old, as some French patent specifications of Renault, dated 1902, show. These early attempts were soon abandoned, however. Only during the late war, owing to the necessity of building fighting airplanes of light weight and high speed, to enable them to quickly overtake and get above the exploring or bombing machines of the enemy, did the problems of an engine capable of working in rarified atmosphere without a reduction of power, become of prime importance.

Notable experimental studies were undertaken for this purpose in different belligerent countries between 1916 and 1920; in Italy by Anastasi[1], in Germany by Brown Boveri, by Siemens-Schuckert and by Schwade in Erfurt[2], in France by Rateau[3], in the United States by Sherbondy in connection with the De Laval Steam Turbine Company, and by Moss in connection with the General Electric Company[4]. These researches, which related exclusively to the employment of a high speed centrifugal compressor driven either by the crankshaft through a speed increasing gear or by an exhaust gas turbine, directed the attention of engineers to the application of a supercharger in cases where greatly increased power per unit of cylinder volume is needed.

Thus the problem has presented itself in a very enticing form, in particular to designers of racing engines, and has been solved with complete success by the Fiat Company, which in 1922-23 built the first supercharged engines, in which a rotary supercharger driven directly by the crankshaft was employed.*

The splendid results obtained in 1923 by the Italian racing cars in the Grand Prix contests in France and in other European countries opened up new vistas in the motoring field, and the next year most of the important motor firms in Europe and America followed the example of the Fiat and took part in international races with cars fitted with superchargers.

From the first conception of the supercharging idea, i. e., to obviate high speeds, it was only a logical development to employ intake pressures several lb. per sq. in. in excess of atmospheric, and thus fill the cylinders with a considerably greater weight of charge. At the same time the first attempts were undertaken to employ supercharged engines in touring cars[5] and in Diesel engines for marine use[6].

In this study it is intended to show how supercharging affects the power developed by car engines, and in particular by racing car engines, and what are the most important technical and mechanical features of operation of such special engines. The methods developed and the conclusions reached can be then applied also to other types of supercharged engine.

In dealing with the thermal cycles, the graphical method of the entropy diagrams with e and T for the coordinates will be used. By this method the thermal efficiency of the cycle may be easily determined, as a ratio of the calories converted into work, i. e., of the area of the cycle to the total heat of the charge liberated in the cylinder.

To take account, in a convenient manner, of the variation of the specific heat of the gases due to the variations of temperature corresponding to the diverse phases of the cycle, the gas entropy chart of Stodola[7] will be used, which answers well for the purpose.

Furthermore, in order to attain a greater approximation to the real conditions, the ideal diagram, composed of

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[1] A. Anastasi: Il motore per i voli d'alta quota. Atti dell' Associazione Italiana di Aerotecnica, volume I, 1920-21.

[2] W. G. Noack: Flugzeuggeblase, Zeitschrift des Vereines deutscher Ingenieure. Bd. 63 Nr. 41; Oct. 11, 1919, page 995.

[3] A. Rateau: Theorie generale du turbo—compresseur pour moteur d'avions. Comptes rendus de l'Academie de France: Vol. 174, June 12, 1922. Los plus grandes vitesses possibles en aviation, et l'omploi du turbo-compresseur; Extrait du Congres Scientifique International, Liege, 1922.—The use of turbo-compressor for attaining the greatest speeds in aviation. Proceeding of the Institution of Mechanical Engineers, Paris and London, 1922.

[4] Leslie V. Spencer: Maintaining constant pressure before the carburetors of aero engines regardless of the altitude, Aerial Age Weekly; April 14, 1919, page 244.

[5] The Autocar, Nov., 1924, page 1103. "Supercharging at Low Speeds."

[6] The Motor Ship, Nov., 1924, page 204. "Superchargers for Four-Cycle Engines."

[7] A. Stodola: Dampf-und Gasturbinen, 5. Edition, Berlin, J. Springer, 1922, page 974.

*The first application of a supercharger to racing engines was apparently made in this country, as pointed out in a letter by Mr. Short in our issue of April 22.—Editor.

two adiabatics and two constant volume lines, will be modified; the corners of the diagram corresponding to the beginning of expansion, to the advanced ignition and to the commencement of exhaust will be rounded off; the heat conducted away by the cylinder walls and by the cooling water also will be taken into account, and the assumption made in accordance with experimental data obtained on similar engines, that 80-85 per cent of the liberated heat at the end of compression remains actually in the gas while the rest can be considered as given up to the cylinder walls.

On the other hand, the differencess due to the mixture not being a perfect gas will be neglected, and the effect of dissociation at the high temperatures of the burnt gases also will not be considered in the calculation. This simplification appears justified on several grounds.

THIS article by Dr. Treves introduces a method of dealing with thermodynamic problems with which probably few of our readers are familiar. This is by the entropy chart of Stodola, an authority on steam turbines.

After the development of equations of a general nature, these are applied to the specific example of a racing engine of the 2-liter or 122 cu. in. class, and the indicated horsepowers required for driving supercharger, net brake horsepower and brake thermal efficiencies for supercharges of 20, 40 and 60 per cent respectively are calculated.

The article should prove of interest to the engineering reader, both on account of the method of analysis employed and on account of the results obtained by it. The premises on which the analysis is based are clearly pointed out. In connection with the example it should be borne in mind, however, that this is a racing engine and not a stock passenger car engine.

The gas entropy chart of Stodola refers to one kilogram-molecule of gas, i. e., an amount of gas weighing as many kilograms as there are units in its molecular weight. The advantage of choosing this unit lies in the fact that by these means the formulae then have the same form for all gases.

Let m be the molecular weight of a gas; R, its constant referring to one kilogram; v, its specific volume in m^3/kg. at a pressure p and absolute temperature T, then

$$R_m = Const. = R = 845$$
$$v_m = Const. = B$$

for all gases[8]. That is to say, the constant of the gas and the volume in cubic meters, both referring to one molecule-kilogram, are the same for all gases, provided the conditions (i. e., pressure and temperature) are equal. For example, at 300 deg. C. abs. and at atmospheric pressure, i. e., for the base point A of the chart,

$$B = 25.35\ m^3$$

For a mixture of different gases, if v_i, m_i G_i are, respectively, the specific volume, the molecular weight and the weight contained in the mixture of one of the gases then

$$B = \Sigma\, m_i v_i = m\, V;$$

the number of molecule kilograms of one component gas is

$$n_i = \frac{G_i}{m_i}$$

and the apparent molecular weight of the mixture will be

$$m = \frac{\Sigma\, n_i m_i}{\Sigma\, n_i}$$

The specific heat at constant volume C_v for a molecule-kilogram can be obtained by multiplying the specific heat of one kilogram of gas by the molecular weight m; according to the latest researches relating to the variation of the specific heat[9].

$$C_v = a + bT$$

where

$$a = 4.67$$

for all gases, while b is a constant different for every gas; for a mixture of different gases:

$$b = \frac{\Sigma\, b_i n_i}{\Sigma\, n_i}$$

The specific heat for one molecule-kilogram of gas at constant pressure can be written:

$$C_p = C_v + A\,R = C_v + 2$$

In consequence of the variation of the specific heat with the temperature the axis of ordinates of the entropy chart of Stodola is in general not perpendicular to the axis of abscissas, but is inclined thereto. The angle of inclination varies for different gases according to the value of constant b of the specific heat.

The direction of the ordinate axis for a given gas or mixture can be determined by connecting the base point A of the chart with the point D, representing—on the scale on the upper part of Fig 1—the value of b for the given gas or mixture. The adiabatic lines will be parallel with the axis of ordinates thus determined; the isothermals will be horizontal; the lines of constant pressure and of constant volume are shown in the table by full and dotted lines, respectively.

In order to facilitate finding the amount of heat, there are marked on the chart the curves of heat, which form a bundle of parabolas having their origin on a line through the point O and ending on the scale of the quantity of heat corresponding to the different values of constant b. With the aid of these lines the amount of heat involved in a transformation at constant volume or constant pressure can be easily found by taking the difference of the two horizontal sections between the straight lines OB or OC, and the curve of the heat for the gas or mixture considered. In a similar manner the heat equivalent to the external work in an adiabatic change can be determined, bearing in mind that this heat is equivalent to the internal work of the transformation.

In drawing the diagram, for the sake of convenience, the assumption will be made, that at the end of the compression the combustible mixture is instantaneously transformed into burnt gas and the heat contained in the fuel will be considered as supplied to the gases from an external source. This heat, which may be called (after J. Thomsen[10]) "detonating heat" can be determined by dividing the heat value of 1 Kg. of fuel by the number of molecule-kilogrammes of the burnt gases corresponding to that amount of fuel; therefore this "detonating heat" depends essentially on the amount of air in the mixture, i.e., on its greater or lesser richness. Gasoline consists on the average of 85 per cent by weight of carbon and 15 per cent of hydrogen; its heat value is 10,200 cal./kg. For the complete combustion of 1 kg. of gasoline, 3.47 Kg. of oxygen is theoretically necessary, which quantity is

[8] P. Ostertag: Die Entropie—Diagramme der Verbrennungsmotoren einschliesslich der Gasturbine, Berlin, J. Springer, 1912, page 2.

[9] Schule; Techn. Thermodynamik, 3rd edition, Vol. I, page 73; also Zeitschrift des Vereines deutscher Ingenieure, 1912, page 78.

[10] P. Ostertag, op. cit., page 6.

contained in 15.07 kg. of air, the difference in weight being accounted for by the nitrogen. The weight of the burnt gases will be 16.07 kg. which is made up of 3.12 kg. carbonic acid, 1.35 kg. of super-heated steam and 11.60 kg. of nitrogen.

From these figures the respective numbers of molecule-kilograms can be calculated:

For the carbonic acid ..... $n_1=3.12/44=0.071$
For the super-heated steam .. $n_2=1.35/18=0.075$
For the nitrogen ..... $n_3=11.6/28=0.415$

making a total of $N=0.561$ molecule-kilograms for 1 kg. of gasoline. The detonating heat will be

$$Q = \frac{10200}{0.561} = 18200 \text{ cal./mol-kg. (appr.)}$$

Proceeding further to the determination of the constant b of the specific heat, we have:

For one mol.-kg. of CO_2 ..... $b = 0.00568$
For one mol.-kg. of H_2O ..... $b_2=0.075$
For one mol.-kg. of N_2 ..... $b_3=0.00106$

therefore for 1 mol.-kg. of burnt gases

$$b = \frac{\Sigma\, b_i\, n_i}{\Sigma\, n_i} = \frac{0.0011515}{0.561} = 0.00206$$

In a similar manner the values for an air supply in excess of the theoretically necessary amount can be calculated:

Proportional air supply	1.5	1.75	.2	2.5
Detonating heat in cal./mol.-kg.	12100	10450	9250	7500
Resultant constant b* ..	0.00175	0.00165	0.00158	0.00140

*Of the specific heat for 1 mol.-kg. of the burnt gases.

The theoretical value for the detonating heat Qdt thus found has to undergo a two-fold correction before it can be used for determining the effective diagram. In the first place it has to be divided by a co-efficient of 1.08—1.15 in order to allow for the presence within the combustion chamber of the burnt gases, which thus augment the number of mol.-kgs. per kg. of gasoline. Secondly, it has to be multiplied, as already mentioned above, by a co-efficient 0.8—0.85 in order to allow for the amount of heat given up to the cylinder walls.

Of course, these co-efficients must be suitably chosen in each particular case, taking into consideration the characteristic features of the engine planned. The value thus arrived at can be called the *"effective detonating heat"* and will be indicated in the following by Q_d.

Drawing the Chart

Let us proceed now to drawing the diagram of the cycle. The characteristic data of the physical state of one molecule-kilogram of mixture at the beginning of compression are p_1, B_1 T_1, corresponding to the first apex A of the diagram. For non-super-charged engine p_1 and T_1 correspond to atmospheric conditions[11] and B_1 can be calculated from the equation of the physical state of the gas. For super-charged engines the physical state of the mixture at the outlet of the super-charger or cooler has to be considered, taking into account also the influence of the inlet manifold and of the carburetor. For an engine to be designed the following mean values may be assumed as holding good for about 6000 r.p.m., actually obtainable in racing engines:

$p_1 = 1.5 - 1.7$ *atm. abs.*; $T_1 = 310 - 340°$ *Cent. abs.*

Then from the known equation

$$10000\, p_1\, B_1 = RT_1.$$

[11] Stodola recommends using the following values: $p_1 = 1$ atm. abs.; $T_1 = 300$ deg. Cent. abs.; $B_1 = 25.35$ m³/mol.-kg. corresponding to the base point A of the entropy chart.

the specific volume B_1 corresponding to the beginning of the compression for a mol.-kg. of mixture can be calculated.

Assuming now that the compression line is adiabatic and using the constant b of air ($b=0.00106$) for the fresh mixture, the compression line can be drawn parallel with the axis of ordinate corresponding to that constant, as set forth above.

Taking now the compression ratio

ρ = *total volume of cylinder/compression volume*

the second apex A_2 of the diagram can be found by extending the compression line until it intersects with the line of constant volume:

$$B_2 = \frac{B_1}{\rho}.$$

This method of construction pre-supposes an instantaneous combustion. The effect of ignition advance, which may amount to 70-80 deg., has to be taken into account, however. For this reason the compression and explosion lines are connected by a curve starting from that point of the compression line which corresponds to the point of ignition. This point can be easily found by drawing the compression line in a *(p, v)* diagram. The compression ratio is fixed when starting the new design. Logically this should be less in super-charged engines than in those without super-charging, and one might even expect that it should be reduced to such a degree that the usual compression pressures would not be exceeded. However, thanks to the use of special heat-resisting types of spark plugs, of special fuels with a base of benzol or alcohol, and to the use, in small quantities, of antidetonating dopes[12], it has been possible to employ the same high compression ratios of 6—7.5 as are being used in non-supercharged racing engines.

The third apex A_3 of the cycle lies on the line of constant volume—

$$B_2 = const.$$

In order to find the position of A_3 it must be considered that it is the effective detonating heat Q_d which causes the gas to change its state from A_2 to A_3. Therefore, if the heat curve OL is laid out for that value of b which corresponds to the burnt gases with the particular surplus of air considered,

$$EF - GH = HI = C_v\left(T_{A_3} - T_{A_2}\right) = Q_d$$

Thus the point A_3 can be obtained by measuring from point H a distance HI corresponding to Q_d on the heat scale and drawing from the point thus found a vertical line until at F it intersects with the heat curve. Where the horizontal drawn from F intersects the line of constant volume $B_2=const.$ is the point A_3 sought.

As the heat given up to the cooling water has already been allowed for, and, furthermore, as instantaneous combustion has been assumed, the subsequent expansion will be of necessity adiabatic and will be represented by a straight line A_3 A_4 parallel to the ordinate axis corresponding to the value b of the burnt gases. Bearing in mind that a transformation A_4 A_1 at constant volume would bring the gases back to their initial state, it will be clear that the amount of heat to be taken away from the burnt gases in order to obtain such transformation will exactly correspond to the heat loss due to the exhaust gases:

$$Q_s = C_v\left(T_{A_4} - T_{A_1}\right) = MN - PQ = RN$$

[12] The Automobile Engineer, Oct., 1924, page 308.

In order to complete the diagram the two lines A_3 A_4 and A_1 A_2 have to be connected by a curve which has its origin at a point corresponding to the beginning of the exhaust and which can be easily found by referring to the *(p, v)* diagram and transferring the curve point by point to the entropy chart. In a similar manner a short curve connects the lines of explosion and of expansion.

It is to be noted that by laying out the cycle as described, and especially if high inlet pressures and a small surplus of air above the amount theoretically required for complete combustion prevail, very high values for the temperature and pressure are arrived at. This result is due to the assumption that all of the detonating heat is supplied instantaneously, at the top dead center. This assumption does not exactly fit the facts, especially at very high speeds. In this case it is probable that the combustion is extended over the whole period of expansion, thus reducing the value of the polytropic exponent considerably below the value $\gamma=1.41$, and raising the curve well above the adiabatic line. However, the error committed by assuming an instantaneous combustion is approximately compensated by assuming an adiabatic expansion and the results thus obtained agree fairly well with the values found by experiments.

Having thus determined the actual cycle, the thermal efficiency of the engine can be easily found. As is well known, the thermal efficiency is the ratio of the heat utilized, Q_u, which corresponds to the area of the cycle, (taking of course the scale of the diagram into consideration), to the heat expended:

$$\eta_t = \frac{Q_u}{Q_{at}},$$

The heat equivalent of the work obtainable from one cubic meter of gas in the physical state corresponding to point A_1 will be Q_u/B_1.

So far the work corresponding to the difference between the pressures of the exhaust (which takes place at atmospheric pressure, p_o) and of the admission (which takes place at the pressure p_1 of the super-charger) has not been taken into account. The thermal equivalent of this supplementary work L_s for one cubic meter of gas will be—

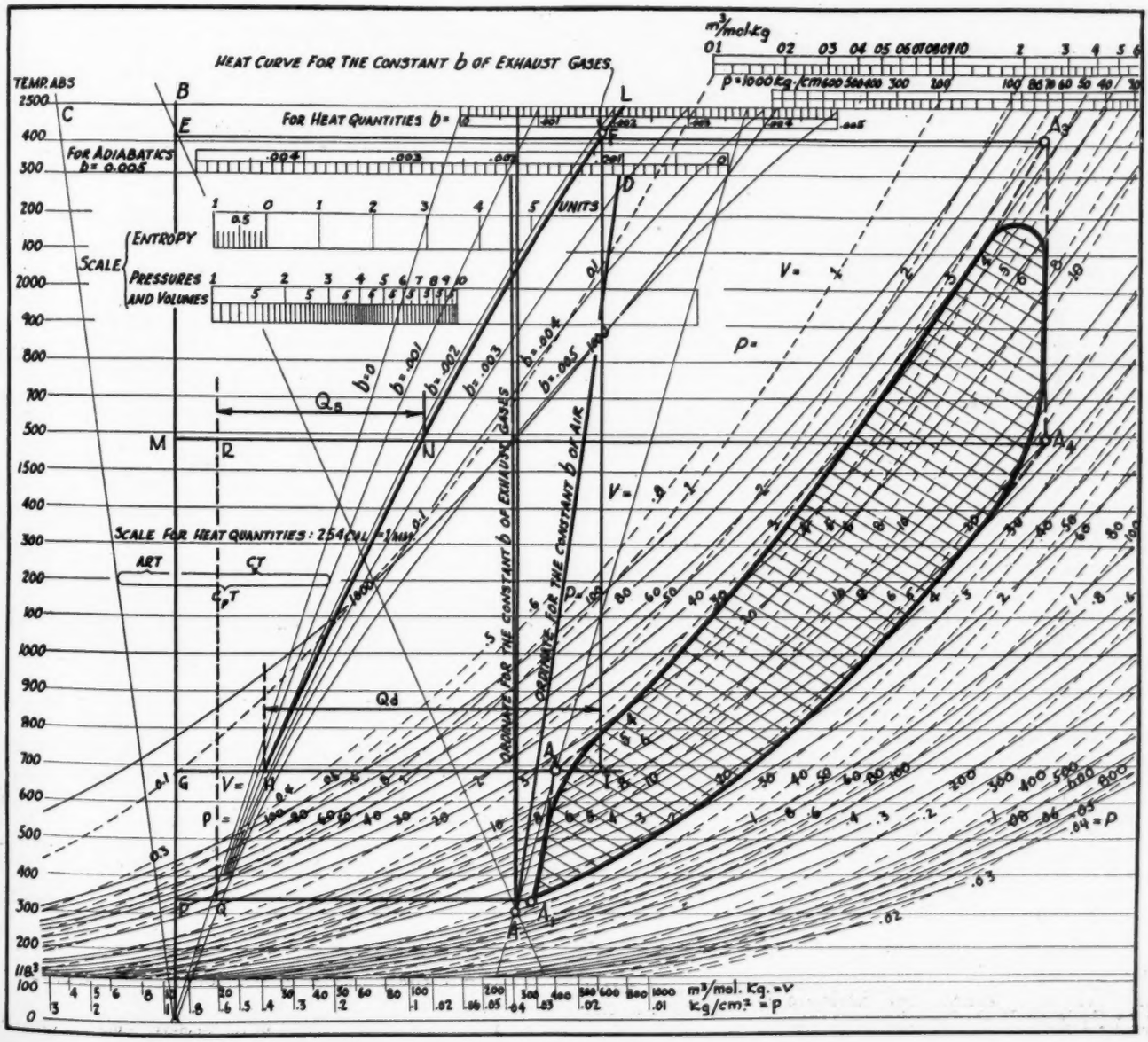
$$A\,L_s = A\,(p_1 - p_o)$$

The exhaust gas displaced in a four-stroke engine is

$$Vni/2 \times 60 \text{ cubic meters per second}$$

where V is the total volume of fresh gas entering one

Fig. 1. Stodola Entropy Chart for Internal Combustion Engine


HEAT CURVE FOR THE CONSTANT b OF EXHAUST GASES
TEMP ABS
FOR HEAT QUANTITIES b=
FOR ADIABATICS b=0.005
SCALE
ENTROPY
PRESSURES AND VOLUMES
UNITS
SCALE FOR HEAT QUANTITIES: 254 CAL = 1 MM
ORDINATE FOR THE CONSTANT b OF EXHAUST GASES
ORDINATE FOR THE CONSTANT b OF AIR
m³/mol.Kg
Kg/cm² = p

cylinder during the inlet stroke; n, the speed in r. p. m., and i, the number of cylinders. Furthermore, if V_p is the swept volume and V_c the compression space for a single cylinder—

$$V = V_p + V_c - V'_c \quad \ldots\ldots\ldots\ldots (1)$$

where V'_c is the volume to which the remaining burnt gases are reduced at the beginning of the admission stroke. At the end of the exhaust stroke these burnt gases are at atmospheric pressure p_o but they are compressed by the fresh charge to the pressure p_1; the transformation takes place approximately polytropically with an exponent k which is rather greater than the adiabatic, owing to the transmission of heat to the gases from the cylinder walls and to the throttling undergone by the mixture at the opening period of the inlet valve; hence

$$V'_c = V_c \left(\frac{p_o}{p_1} \right)^{\frac{1}{k}} \quad \ldots\ldots\ldots\ldots (2)$$

where values of 1.6-1.7 can be used for k[13].

The effective brake horsepower of the engine, including also the power necessary for driving the supercharger, can be now expressed. If η_m is the mechanical efficiency, and p and p_o are expressed in kg./cm², in the case of a four-stroke cycle engine

$$N_e = \frac{427\ ni}{2x60x75} \left\{ \frac{Q_u}{B_1} \left(V_h + V_c \left[1 - \left(\frac{p_o}{p_1} \right)^{\frac{1}{k}} \right] \right) + 10{,}000\ AV_h\ (p_1 - p_o) \right\} \eta_m \quad \ldots\ldots\ldots (3)$$

and in the case of a two-stroke cycle engine:

$$N_e = \frac{427\ ni}{60x75} \left\{ \frac{Q_u}{B_1} \left(V_h + V_c \left[1 - \left(\frac{p_o}{p_1} \right)^{\frac{1}{k}} \right] \right) + 10{,}000\ AV_h\ (p_1 - p_o) \right\} \eta_m \quad \ldots\ldots\ldots (3_a).$$

In order to check the result obtained, the value of N_e could also be calculated by drawing the theoretical cycle, i.e., *without* correcting the detonating heat and *without* rounding off the apices. In this case, evidently,

$$Q_u = Q_d - Q_s$$

With the horsepower values N_t thus obtained in the case of four-stroke cycle engines,

$$\eta_v = \frac{N_e}{N_t} = 0.8 - 0.9$$

and in the case of two-stroke cycle engines,

$$\eta_v = \frac{N_e}{N_t} = 0.7 - 0.8$$

The value N_v signifies the ratio between the area of the cycle actually obtained and that of the theoretical cycle. The difference between the two areas decreases with increase in speed, owing to the decrease of the polytropic exponent in comparison with the adiabatic one on the one hand and owing to the reduced amount of heat transmitted to the cooling water occasioned by reduced duration of the explosion and expansion, during which the high temperature gases are in contact with the cooled cylinder walls, on the other hand.

As regards the value to be given to the mechanical efficiency in equations (3) and (3_a) it has to be considered that racing engines always embody a specially high grade of workmanship and are generally fitted with roller bearings, and the rotating masses, crankpins, connecting rods, etc., are carefully balanced for each crankpin; therefore these values are very high, and on the average

$$\eta_m = 0.90 - 0.93$$

From the entropy diagram and its transformation into a p-v diagram, the indicated and effective mean pressures can be conveniently obtained by the usual methods.

It is interesting to compare the diagrams of the cycles hitherto considered with the diagram of an engine without super-charging. In this case, owing to the low value of the volumetric efficiency, the residual gases will occupy a considerable percentage of the cylinder volume. For instance, if the volumetric efficiency be 0.70, then the residual gases will occupy 30 per cent of the swept volume; therefore in calculating the detonating heat, the number of the molecule-kilograms relating to one kg. of fuel would have to be multiplied by 1.30 which has the same effect as dividing the detonating heat by the same value 1.30. Furthermore, as the admission pressure in equations (3) and (3_a) referring to the effective horsepower, is equal to the atmospheric, the last term of the second member evidently vanishes.

To obtain a complete power balance for a supercharged engine and judge the increase of useful brake power, the power necessary for driving the supercharger must be calculated. In most cases a Roots type rotating compressor having two lobes of the form of an 8 is used; the centrifugal type of compressor is used solely in a few special constructions[14]; piston compressors are not employed, for obvious reasons. The pressure p_c to be supplied by the supercharger differs only slightly from the admission pressure p_1, owing to the short length and favorable design of the inlet manifold. This difference will reach the value of about one-tenth atmosphere only when a cooler is being used. Assuming the compression to be adiabatic, the corresponding temperature T_c can be determined from the equation:

$$T_c = T_o \left(\frac{p_c}{p_o} \right)^{\frac{\gamma - 1}{\gamma}}$$

in which formula $\gamma = 1.41$ is the well-known ratio of specific heat at constant pressure and at constant volume, while T_o and p_o are the temperature and pressure of the atmosphere. These latter values can be taken as 300 deg. Cent. abs. and 1 kg./cm², respectively.

For one kilogram of air the theoretical work of compression will be (in kilogram-meters)

$$427\ c_p \left(T_c - T_o \right) = 427\ c_p\ T_o \left[\left(\frac{p_c}{p_o} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right]$$

where c_p (=0.24) signifies the specific heat of air at constant pressure[15]. Furthermore, remembering that (as explained above) the volume of mixture corresponding to one admission stroke is

$$V = V_h + V_c - V'_c = V_h + V_c \left[1 - \left(\frac{p_o}{p_1} \right)^{\frac{1}{k}} \right],$$

it follows from Dalton's law that V is also the volume of air contained in the mixture. The pressure of this air is slightly less than p_1, owing to the vapor pressure of the fuel. This difference is negligible, however.

It is necessary to supply an excess of air of e per cent owing to the losses in the inlet manifold unions. There-

[13] On the basis of experiments carried out on an Hispano-Suiza airplane engine for the purpose of determining the volumetric efficiency as a function of the exhaust pressure. Kemble recommends the assumption $k = 2$. This value, however, seems to be rather high. (See: E. C. Kemble, The Calculated Performances of Airplanes Equipped With Supercharging Engines; National Advisory Committee for Aeronautics, Washington, 1921).

[14] On the recent racing cars of Duesenberg and Miller in America, for instance.

[15] The exact values for air in the neighborhood of 0 deg. Cent. are:

$$c_p = 0.2387; \quad \gamma = \frac{c_p}{c_v} = 1.4034; \quad \frac{\gamma - 1}{\gamma} = 0.2875$$

J. R. Partington, Proc. of the Royal Society, London, Oct., 1921.

fore, the total volume will be $V\,(1+e)$. Dividing this volume by the specific volume at the end of admission

$$V_1 = \frac{R\,T_1}{10{,}000\,p_1},$$

the weight G of air (in kilograms) is obtained. The coefficient η_{ca} corresponding to the ratio between the actual and the adiabatic compression and also the mechanical efficiency η_{cm} of the blower have to be taken into account. With all these considerations the power necessary for driving the supercharger will be for a four-stroke cycle engine:

$$N_c = \frac{427\,ni}{2x60x75}\,c_p\,T_o\left[\left(\frac{p_c}{p_o}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \frac{\left\{V_h + V_c\left[1-\left(\frac{p_o}{p_1}\right)^{\frac{1}{k}}\right]\right\}(1+e)}{\frac{R\,T_1}{10{,}000\,p_1}} \cdot \frac{1}{\eta_{ca}\,\eta_{cm}} \quad \ldots(4)$$

As a mean value for the excess of air we may assume $e=0.10$ and for the mechanical efficiency of the blower, considering the specially high-grade of workmanship and refinements of design (use of roller bearings, etc.), we may take $\eta_{cm}=0.93$—0.95. On the other hand the value of η_{ca} is rather low and in the Roots type of blower is not higher than $\eta_{ca}=0.55$—0.60, owing to the internal losses, which are converted into heat and raise the temperature of the air. This value η_{ca} expresses also the ratio between the temperature rise corresponding to the adiabatic and that corresponding to the actual transformation of state, i. e.,

$$T' = \frac{T_c - T_o}{\eta_{ca}} + T_o = \frac{T_o}{\eta_{ca}}\left[\left(\frac{p_c}{p_o}\right)^{\frac{\gamma-1}{\gamma}} + \eta_{ca} - 1\right]$$

This considerable rise in temperature explains the need for an air-cooler between the supercharger and the engine, to reduce the temperature and increase the density of the entering charge, if an increase of a few tenths of an atmosphere in the admission pressure is desired.

Subtracting the power N_c necessary for driving the supercharger from the effective power N_e of the engine, the useful power N_u, which is measurable on the brake, can be obtained. Denoting by N_{eo} the power of the engine without supercharge (as calculated above), the percentual power increase due to supercharging will be:

$$100\,\frac{N_u - N_{eo}}{N_{eo}}$$

Finally, the efficiency ϵ of the engine-supercharger unit can be expressed as follows:

$$\epsilon = \frac{N_u}{N_i} = \frac{1}{\frac{Q_{dt}}{B_1}\left\{V_h + V_c\left[1-\left(\frac{p_o}{p_1}\right)^{\frac{1}{k}}\right]\right\}} \left|\left(\frac{Q_u}{B_1}\left\{V_h + V_c\left[1-\left(\frac{p_o}{p_1}\right)^{\frac{1}{k}}\right]\right\} + 10000\,AV_h\,(p_1-p_o)\right)\eta_m\right.$$

$$\left. - c_p\,T_o\left[\left(\frac{p_c}{p_o}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]\frac{\left\{V^h + V_c\left[1-\left(\frac{p_o}{p_1}\right)^{\frac{1}{k}}\right]\right\}(1+e)}{\frac{RT_1}{10000p_1}}\,\frac{1}{\eta_{ca}\,\eta_{cm}}\right| \quad \ldots\ldots(5)$$

In the case of the non-supercharged engine, where

$$p_o = p_1 = p_c$$

equation (5) reduces to the simplified form

$$\epsilon_o = \frac{Q_u}{Q_{dt}}\,\eta_m$$

By way of example, the equations derived will be applied to a four stroke cycle racing engine of the type used by Fiat and Alfa Romeo in 1924 in the Grand Prix of Europe contest at Lyons, France. This type has eight cylinders in line with a swept volume V_p=0.00025 cu. meters for each cylinder, corresponding to a total volume of 2 liters. A compression ratio of ρ=6.5 will be assumed, corresponding to a combustion space

$$V_c = \frac{V_h}{\rho - 1} = 0.0000454\ m^3$$

The following values will be used:

$$n = 6000;\ k = 1.65;\ e = 0.10$$

$$\eta_m = 0.91;\ \eta_{ca} = 0.58;\ \eta_{cm} = 0.94$$

$$T_o = 300\ \text{deg. Cent. abs.};\ p_o = 1\ \text{atm. abs.};$$
$$Q_{dt} = 18200\ \text{cal/mol.-kg.}$$

A pressure drop of 0.1 atm. between the outlet pressure of the supercharger and the initial pressure of the compression stroke will be assumed. On the basis of the well-known researches of Mallard and Le Chatelier, and of E. Meyer, the value R=31.83 will be used for the fresh mixture and R=30.61 for the burnt gases. Taking the effect of the air-cooler into account, the temperature of the mixture at the beginning of compression wil be assumed to be

$$T_f = 303\ \text{deg. C. abs., corresponding to 30 deg. C. or 86 deg. F.,}$$

and that of the residual burnt gases,

$$T_r = 873\ \text{deg. C. abs., corresponding to 600 deg. C. or 1,112 deg. F.}$$

In fact, rather higher temperatures (820—840 deg. C.) will be found by direct measurement by means of a thermo-couple placed in the exhaust pipe close to the cylinder. And even this value can be considered as below the actual value at the beginning of the exhaust stroke, bearing in mind that there is only one exhaust stroke in four strokes and therefore the temperature measured is some mean value of the four strokes. Nevertheless T_r is actually much less, because the residual gases give up heat to the cylinder walls during the whole of the exhaust stroke, so that the value mentioned above can be considered as fairly correct. On the other hand, even a considerable difference in the value of T_r would not cause any great variation in the result of the calculation.

Assuming, furthermore, that the fresh charge mixes with the residual products without any heat being lost to or received from the outside, the following thermodynamic equation may be written—

$$\frac{V_c\left(\frac{p_o}{p_1}\right)^{\frac{1}{k}}}{T_r} + \frac{V_h + V_c\left[1-\left(\frac{p_o}{p_1}\right)^{\frac{1}{k}}\right]}{T_f} = \frac{V_h + V_c}{T_1}$$

from which T_1 may be calculated, by varying the supercharging pressure from 1.2 to 1.7 atm. abs. the resulting values for T_1 will be between 331 deg. and 327 deg. Cent. abs. It is therefore safe to assume for the sake of simplicity that

$$T_1 = 333\ \text{deg. Cent. abs. corresponding to 60 deg. C.}$$

for all pressures. In this value the heat received from the cylinder walls during the admission stroke is taken into consideration also.

If, then, we draw the diagram of the cycle on the entropy chart (Fig. 3) and apply the calculations previously described, the values tabulated in Table 1 are obtained.

Table 1

Admission pressure — p_1 atm. abs.	1.2	1.4	1.6
Heat utilized per cycle—Qu cal/mol.-kg.	5100	5220	5290
Specific volume of charge at the beginning of the compression stroke —B_1 cu. m./mol. kg.	23.2	20.1	17.56
Total volume of fresh charge per admission stroke — $V = V_h + V_c \left[1 - \left(\frac{p_o}{p_1} \right)^{\frac{1}{k}} \right] m^3$	0.00025475	0.00025837	0.0002612
Effective power of the engine —Ne HP.	117.5	144.5	170.8
Pressure at supercharger outlet — Pc atm. abs.	1.3	1.5	1.7
Temperature at supercharger outlet — T'c abs.	340	364	385
Power absorbed by supercharger — Nc HP.	7.98	14.65	23.15
Useful power — Nu HP.	109.5	127.8	147.65
Brake thermal efficiency of the engine supercharger unit: e	0.2415	0.2400	0.2395

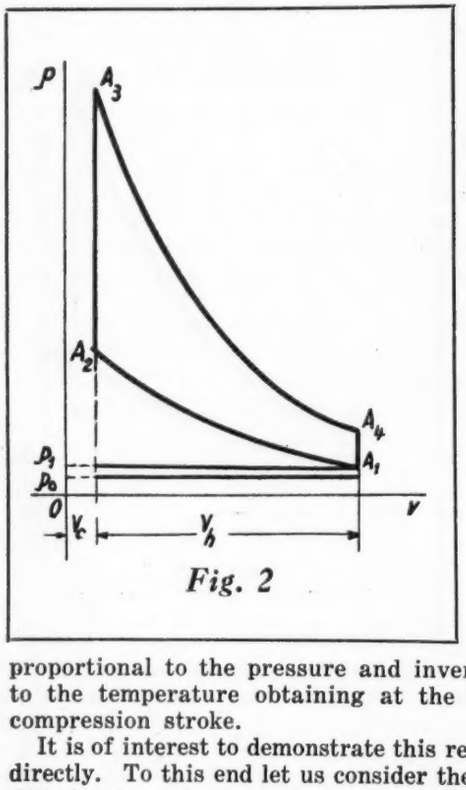


Fig. 2

On the basis of the preceding calculation, and in particular of the examples worked out, it can be stated, that in a supercharged engine at a given angular speed the power corresponding to the theoretical cycle and approximately also the indicated power is directly proportional to the density of the air at the beginning of the compression stroke; in other terms, it is directly proportional to the pressure and inversely proportional to the temperature obtaining at the beginning of the compression stroke.

It is of interest to demonstrate this remarkable property directly. To this end let us consider the cycle $A_1 A_2 A_3 A_4$ (Fig. 2) formed of two lines of constant volume and two polytropics of equal exponent and let us suppose that the density

$$\gamma_1 = \frac{p_1}{\mathrm{R} T_1}$$

changes at point A_1 to

$$\gamma'_1 = \frac{\alpha p_1}{\mathrm{R} \beta T_1}$$

corresponding to a new admission pressure $p_1 = \alpha p_1$ and to a new admission temperature $T_1 = \beta T_1$. As a consequence all the points of the compression polytropic $A_1 A_2$

$$p_1 (V_c + V_h)^n = p_2 V_c^n$$

will be multiplied by the factor α, and thus

$$p'_2 = p'_1 \rho^n = \alpha p_1 \rho^n = \alpha p_2$$

where ρ is the compression ratio. Furthermore, the temperature at A_2 will be:

$$T'_2 = T'_1 \rho^{n-1} = \beta T_1 \rho^{n-1} = \beta T_2$$

As the calorific value H (=10200 cal./kg.) is practically independent of both the pressure and the temperature, the temperature rise ΔT at the moment of explosion will not be altered, and thus we have —

$$\frac{p_3 - p_2}{p_2} = \frac{\Delta T}{T_2} \quad ; \quad \frac{p'_3 - p'_2}{p'_2} = \frac{\Delta T}{T'_2}$$

from which it follows that

$$p'_3 - p'_2 = \frac{\alpha}{\beta} (p_3 - p_2)$$

The equation of the expansion line is

$$p V^n = \left[\alpha p_2 + \frac{\alpha}{\beta} (p_3 - p_2) \right] V_c^n ,$$

in which p and V are the coordinates of a point of the curve.

The equation of the compression line being —

$$p V^n = \alpha p_2 V_c^n ,$$

the difference between the ordinates of the two polytropics corresponding to the same abscissae will be —

$$\Delta p = \left[\alpha p_2 + \frac{\alpha}{\beta} (p_3 - p_2) - \alpha p_2 \right] \left(\frac{V_c}{V} \right)^n$$

$$= \frac{\alpha}{\beta} (p_3 - p_2) \left(\frac{V_c}{V} \right)^n$$

For the area of the cycle we have —

$$A = \int_{V_c}^{V_c + V_h} \Delta p \, dV = \frac{\alpha}{\beta} (p_3 - p_2) V_c^n \int_{V_c}^{V_c + V_h} \frac{dV}{V^n}$$

$$= \frac{\alpha}{\beta} (p_3 - p_2) V_c^n \frac{1}{1-n} \left[(V_c + V_h)^{1-n} - V_c^{1-n} \right]$$

This shows that the ratio of the area of the modified cycle to that of the original is the same as the ratio of the respective specific gravities,

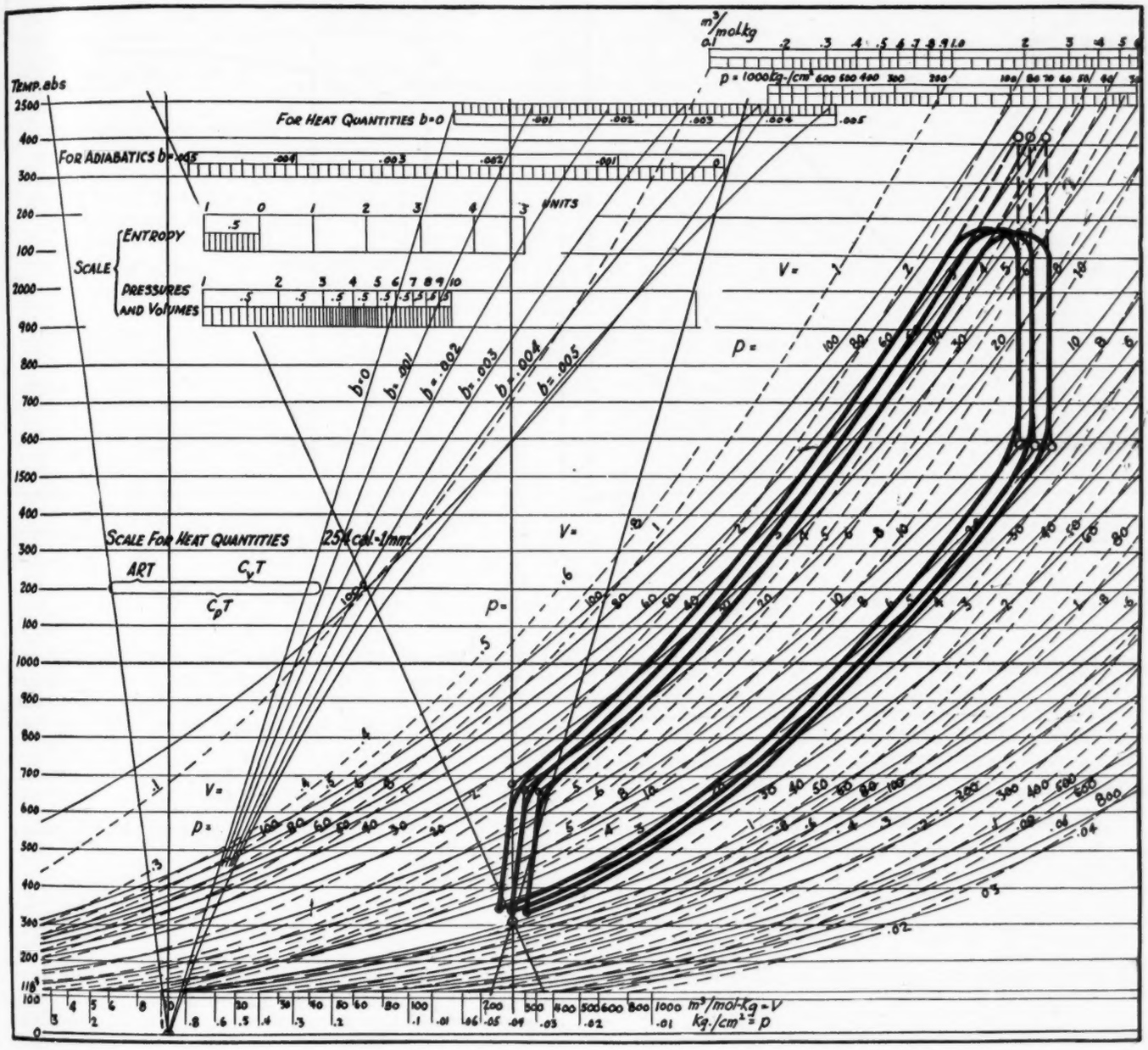
$$\gamma_1 \text{ and } \gamma'_1, \text{ i. e., } \frac{\alpha}{\beta} .$$

Considering further that the area between the two lines of constant pressure, p_o and p_1 varies in proportion to the pressure p_1[16], and also that the rounding off occurring in the actual cycles does not impair the validity of reasoning, it can be stated that the area of the cycle is, with a fair approximation, directly proportional to the density of air at the beginning of the compression stroke. This law of proportionality, which holds essentially good also for the indicated power, has to be modified, if the effective power, *i.e.*, the power measurable by the brake (the supercharger having an independent drive) has to be considered.

In fact, the resistances of the engine are for one-third proportional to the indicated mean effective pressure, and thus to the density of the charge at the beginning of compression, and for two-thirds independent of that pressure.

[16] Bearing in mind the small dimension of this area in comparison with the total area of the cycle, it can be assumed, without any considerable error, that it varies in proportion to the density.

Fig. 3. Entropy Chart for Three Degrees of Supercharging



To the first class belong the frictional losses of the pistons and of the main and connecting rod bearings; to the second the power absorbed by the camshaft, water pump, magneto, etc. The two classes of loss in a non-supercharged engine can be expressed by

$$N_{r1} = \xi_1 N_{io} \text{ and } N_{r2} = \xi_2 N_{io},$$

where N_{io} is the indicated power of the non-supercharged engine and ξ_1 and ξ_2 are two factors less than unity. Of these losses, as said above, the first varies in direct proportion to the density of the charge when the engine is supercharged, while the second is independent thereof.

If the ratio of the density of the air at the beginning of the compression stroke in the same engine with and without supercharging is $\delta > 1$, the ratio of the effective power of a supercharged and a non-supercharged engine will be:

$$r = \frac{N_e}{N_{eo}} = \frac{\delta N_{io} - \delta N_{r1} - N_{r2}}{N_{io} - N_{r1} - N_{r2}} = \delta \frac{1 - \xi_1 - \xi_2/\delta}{1 - \xi_1 - \xi_2},$$

and in a similar manner the ratio of the respective mechanical efficiencies can be expressed by

$$s = \frac{\eta_m}{\eta_{mo}} = \frac{N_e}{\delta N_i} \frac{N_i}{N_{eo}} = \frac{1 - \xi_1 - \xi_2/\delta}{1 - \xi_1 - \xi_2},$$

from which the following equation may be written —

$$r = s\,\delta.$$

On the basis of the foregoing, it can be stated that in a supercharged engine, other conditions being equal (i.e. at the same speed and with the same richness of mixture) the effective power and the mechanical efficiency of the engine (excluding the supercharger) vary according to a linear ordinary and to a linear fractional function of the density of the charge at the beginning of the compression. Accordingly, the ratio of the effective powers of the engine with and without supercharging can be represented as a function of the density by a straight line (Fig. 4.), while the ratio of the corresponding efficiencies can be represented by an equilateral hyperbola (Fig. 5) which with increasing δ approaches assymptotically to the value

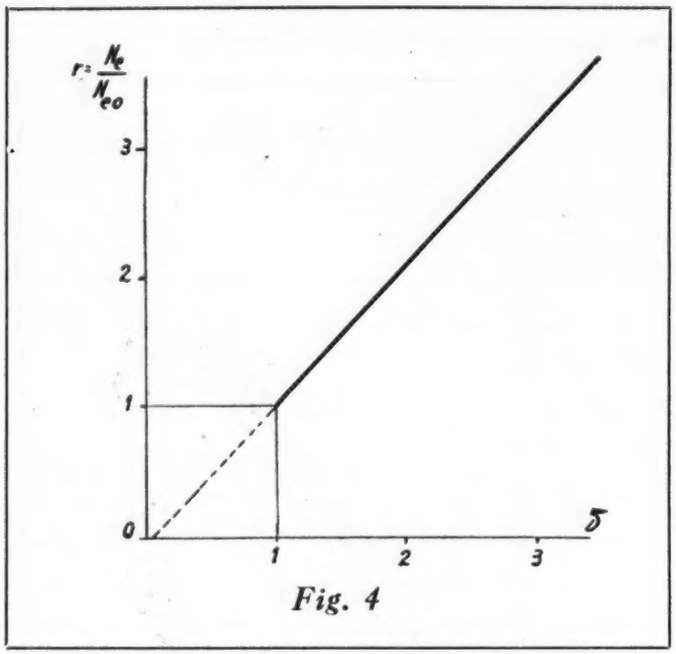


Fig. 4

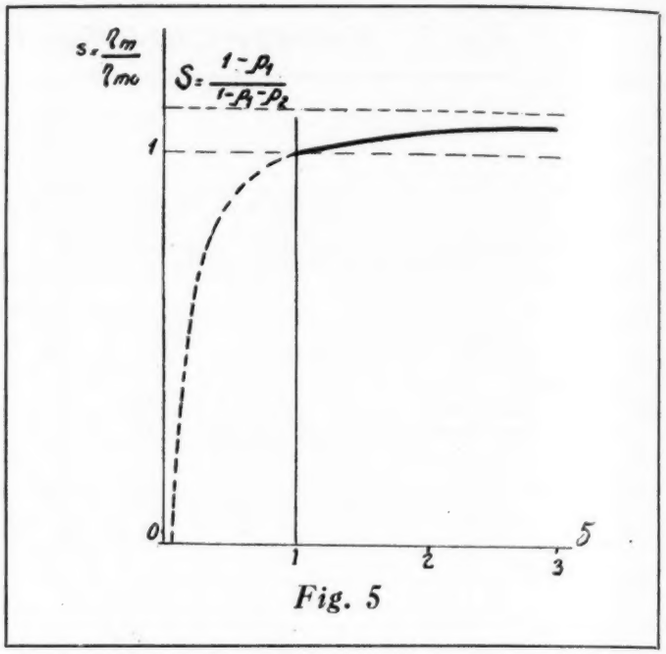


Fig. 5

$$S = \frac{\eta_m}{\eta_{mo}} = \frac{1 - \xi_1}{1 - \xi_1 - \xi_2}.$$

For instance, when

$$\xi_1 = 0.03 \text{ and } \xi_2 = 0.06$$

the value S=1.066 would be obtained.

For each fixed value of δ, all other conditions being equal, one of the two ratios mentioned is a linear function of the other.

If, for the sake of simplicity, we assume that the power absorbed by the frictional losses is constant, i.e.,

$$N_{r1} = O,\ N_{r2} = N_r = (1 - \eta_{mo})\ N_{io},$$

the effective power of the supercharged engine will be:

$$N_e = \delta\ N_{io} - (\ 1 - \eta_{mo}\ )\ N_{io}$$
$$= \eta_{mo} \left(1 + \frac{\delta - 1}{\eta_{mo}} \right) N_{io},$$

while the mechanical efficiency can be expressed by

$$\eta_m = \frac{N_e}{\delta N_{io}} = \eta_{mo} \frac{1 + \frac{\delta - 1}{\eta_{mo}}}{\delta} = \frac{\eta_{mo} + \delta - 1}{\delta}$$

We shall now consider the influence of supercharging on the mixture ratio and the fuel consumption. For the sake of simplicity these considerations will be based on the elementary theory of carburation, i.e., the effects of friction and inertia on the flow of air and fuel will be neglected. It is well known that in this case the theoretical velocities in the venturi throat can be respectively expressed by

$$v_a = \sqrt{2g H_a} \text{ for air and } v_b = \sqrt{2g H_b} \text{ for gasoline,}$$

in which equations H_a and H_b are the pressure differences which produce the flow, expressed in heights of column of each fluid, and g is the acceleration of gravity. Using the symbols d_a and d_b for the densities of the two fluids,

$$H_a\, d_a = H_b\, d_b$$

and, therefore,

$$\frac{v_a}{v_b} = \sqrt{\frac{H_a}{H_b}} = \sqrt{\frac{d_b}{d_a}}.$$

The ratio in weight of fuel to air, i.e., the richness of the mixture can be expressed as follows:

$$R = \frac{s_b\, v_b\, d_b}{s_a\, v_a\, d_a} = \frac{s_b}{s_a} \sqrt{\frac{d_b}{d_a}},$$

where s_b and s_a are the areas of the orifices for fuel and air, respectively. If it is assumed that the temperature (and thus also the density d_b) of the fuel is practically constant under the different conditions of operation, then

$$R = R_o \sqrt{\frac{d_{ao}}{d_a}},$$

where the index o refers to the non-supercharged engine. In other terms,

$$R = R_o \sqrt{\frac{p_{ao}}{p_a} \frac{T_a}{T_{ao}}},$$

These equations can be summed up in the following words: The richness of the mixture is inversely proportional to the square root of the density of the compressed air entering the carburetor; i.e., it varies in inverse proportion with the square root of the pressure and in direct proportion to the square root of the temperature beyond the cooler. It is always assumed that the air cooler is interposed between the supercharger and the carburetor in order to eliminate the danger of an explosion in an air cooler full of explosive mixture. On the other hand, other things being equal, the weight of air flowing through the carburetor in unit time is in direct proportion to its density, from which fact it follows that the consumption of fuel per hour varies as the square root of the density of the compressed air entering the carburetor.

Finally, assuming that the density of the mixture at the beginning of compression is approximately directly proportional to the density of the air entering the carburetor, it follows that the brake thermal efficiency of the engine and its specific fuel consumption (referring to one horse power and one hour), vary in a direct or inverse ratio with an algebraic (but non-rational) function of the type:

$$f(d_a) = a\, d_a^{\frac{1}{2}} + b\, d_a^{-\frac{1}{2}},$$

where a and b are numerical constants depending on the power losses and on the charcteristics of the mixture.

The foregoing brief analysis of the characteristics of supercharged engines and their dependence upon the physical state of the entering compressed air, shows the great influence of the air temperature. If a supercharge of more than two or three-tenths of atmosphere is desired, it is often necessary to employ an air cooler. These conclusions are fully corroborated by experimental studies carried out on racing and aircraft engines in recent years. Thus Devillers[17] referring to experiments carried out during the war, states that by raising the admission pressure above that of the exhaust, an increase of power in direct proportion to that difference of pressure is obtained, which is approximatively independent of the engine speed; futhermore, the increase in admission pressure improves the volumetric efficiency by reducing the weight of the burnt products remaining in the cylinder at the end of the exhaust stroke and thus reducing the fuel consumption per horsepower-hour. A further gain in thermal efficiency and a reduction in the heat lost to the cooling water also result.

Similar conclusions were arrived at from the results of tests carried out on two Hispano-Suiza aircraft engines of 150 and 180 hp., in the Altitude Laboratory of the Bureau of Standards, Washington[18], and also from other researches undertaken by the Royal Aircraft Establishment[19] on an experimental engine of 235 hp. running at 1700 r.p.m. In these experiments the influence of the admission temperature (keeping the admission pressure constant) was made the subject of a particular study. It could be distinctly observed that the power loss caused by an increase of the admission temperature is essentially dependent on the engine speed and directly proportional to the temperature increase, and amounts to about 0.1 per cent per deg. Cent. up to 40 deg. Furthermore, keeping the setting of the carburetor constant, the specific fuel consumption varied as the admission temperature, increasing approximately 1/6 per cent per degree Cent.[20]

From the foregoing considerations of the characteristics of supercharged engines the advantages and inconveniences inherent in this method of operating motor car engines are clear. The same result as with a supercharger cannot be obtained (as might possibly be supposed) by an increase in the compression ratio. This would not increase the volumetric efficiency, but it would increase the temperature of the charge at the end of the compression stroke, with consequent greater liability to pre-ignition, thus rendering the engine more difficult to operate.

All such difficulties can be eliminated in a supercharged engine by fitting an air cooler between the supercharger and the engine. It is also easy to disprove the contention that it would be an easier and a safer plan to increase the displacement of the engine, rather than to resort to supercharging. The answer to this is that at a given speed, and particularly at the high speeds which are now common even in stock engines, the reciprocating and rotating masses are greater in a non-supercharged engine, which implies more wear and tear on the various parts.

Of course, the argument that the greater difficulty of operating such engines, and also the low efficiency of the supercharger, which, as pointed out, hardly exceeds 50 per cent, would militate against its general adoption, cannot be entirely dismissed. Nevertheless, in view of the rapid progress of motor car engineering, and particularly racing car engineering, it can be set forth with assurance that supercharging, which already finds wide application in terrestrial, marine and aerial locomotion, constitutes an important step forward in the development of the internal combustion engine.

[17] R. Devillers: Le Moteur a Explosion, Vol. II, Paris, Dunod, 1921.
[18] The Automobile Engineer, Oct., 1919, p. 342, and Feb., 1920, p. 65.
[19] R. Devillers: *Ibidem*, t. II., p. 681.

[20] Ricardo in England in his studies on a small supercharged engine arrives at a similar conclusion. See: Harry R. Ricardo: Some Experiments on Supercharging in a High Speed Engine. The Automobile Engineer, June, 1921, p. 219.

Thermal Efficiency Calculated by Dr. Treves Probably Unobtainable With Ordinary Fuels

SOME of the results arrived at in the foregoing article by Dr. Treves evidently could not be realized practically with present-day fuels, and this applies particularly to the high thermal efficiency which would seem to be attainable from the calculations. It has long been the impression among engineers that supercharging results in decreased brake thermal efficiency. There are two reasons for this belief. If we supercharge an engine we supply it with more charge than corresponds to its displacement, and for a given compression pressure this means a smaller ratio of expansion. As the thermal efficiency depends upon the expansion ratio, this would lead one to expect a lower efficiency. The other reason is that considerable power is required to drive the supercharger.

It is shown by Dr. Treves that more than 23 hp. is required to operate the supercharger when supercharging a 2 liter or 122 cu. in. engine to 1.6 times atmospheric pressure. This power, of course, comes from the engine crankshaft and does not appear in the brake horsepower on which the brake thermal efficiency is based. In spite of this, Dr. Treves obtains substantially the same brake thermal efficiency when working with 1.6 atmosphere supercharge as when working with only 1.2 atmosphere supercharge. This result is based on two assumptions which could not be realized in practice with commercial fuels.

The first of these assumptions relates to the compression ratio. This is taken to be the same in the supercharged as in the normal engines, viz. 6.5. The assumption is justified by the author on the ground that special non-detonating fuel is used in racing engines which will permit of using such a compression. A compression ratio of 6.5 with a supercharge of 1.6 atmosphere corresponds roughly to a compression ratio of 9.2 without supercharge. With ordinary petroleum fuels the compression pressure which can be used is limited by detonation, and if an engine is supercharged the compression ratio will have to be decreased so that the maximum compression pressure reached is no greater than in the case of the non-supercharged engine. But the expansion ratio will then be less and it is this ratio which determines the thermal efficiency.

In the example an expansion ratio of 6.5 is figured with, but if the compression were kept down to the same value as in a non-supercharged engine of this compression ratio then the expansion ratio of the supercharged engine would be only 4.59. Now, the thermal efficiency varies as

$$1-\left(\frac{1}{r}\right)^{0.35}$$

The value of this expression is 0.4805 for a compression ratio of 6.5 and 0.413 for a compression ratio of 4.59. Hence, for

(Continued on page 1022)

Tire Engineering

Giant low pressure tires regarded caused by heat from brakes. drum design also enter

By Walter L. Carver

BUS balloon tires have been the focus of development effort in the various tire plants during the past year. The reason for this concentration is covered in the following brief statement:

Rubber tires are the only units making up the automotive vehicle which are run to destruction regularly within the life period of the chassis and powerplant. The average small car running about 5000 miles per year on tires of 4.40 section wears about 23 lb. of rubber in that period. A modern bus easily shows a mileage figure of 60,000 in a year's service and in that time consumes 1600 lb. of rubber or more. Therefore, as related to the individual vehicle, the bus tire problem is about 70 times as great as that for the small car.

Since the wide adoption of pneumatic tires for bus and other heavy duty service, tires have not been sold in that field as they are to the individual passenger car owner. Large fleet operators have forced sales on a mileage basis or guaranty and the experience has not been pleasant or profitable. Further, the trouble has not been altogether with the tires but is rooted in unintelligent "tire management" and on too close proximity to brakes, which, in the necessity for maintenance of high average speeds, have been applied so frequently as to produce destructive temperatures, particularly at the inside tire of a dual mounting. The latter difficulty also is based somewhat on the existing trend toward low floor height and restricted wheel housings.

In the opinion of tire manufacturers and engineers, the bus balloon tire offers a solution of this transportation problem, provided bus engineers will cooperate in designing the bus chassis and operators will avail themselves of the established knowledge of tire management. Interviews and questionnaires in the tire industry have produced one unanimous answer. The bus balloon tire is the outstanding engineering development in the tire field during the past year.

It should be stated here that the purpose of this article is to deal with engineering developments only as they apply to operation on the automotive vehicle and therefore intersect with the lines of development of vehicular design and operation.

After the bus balloon tire, interest centers in the drop center rim development, although tire manufacturers feel that this is more largely a problem for the wheel and rim manufacturer. In the rubber industry, the development of a tube which will fit either drop or flat base rim with the same size tire was the real problem.

Other points in the program of engineering development in the tire field include cushion tires or fillers for relatively low speed light commercial operation; advance in passenger car balloon tire practice, although no particular changes in design are involved, and some rim development.

In structure and general make-up, the bus balloon tire, which also is fully applicable to the truck industry, bears about the same general relationship to the heavy duty pneumatic tire as does the passenger car balloon tire to its high pressure cousin. This analogy and the general characteristics are portrayed by the following tables which give the details of one tire manufacturer's production of both high and low pressure heavy duty tires. It may be added that such items as the nominal section

Centers Around Bus Balloons

as a partial remedy for troubles "Tire management" and brake into problems of wear.

size, spacing for dual use, etc., are the current standards of the Tire and Rim Association, Inc.

TABLE 1

Dimensions of Truck and Bus Tires

Tire Size	Plies	Rim Preferred	Center Spacing Preferred	Out. Diameter	Sect. Diameter	Axle to Ground Loaded	Maximum Load and Inflation
Truck and Bus Balloon—Heavy Duty							
32x6.75	6	6 in.	8 3/8 in.	34.24 in.	7.02 in.	15.84 in.	1700 lb. at 50 lb.
34x7.50	6	6 in.	8 3/8 in.	35.33 in.	7.48 in.	16.32 in.	2100 lb. at 50 lb.
36x8.25	8	7 in.	10 1/16 in.	36.73 in.	8.44 in.	16.85 in.	2500 lb. at 50 lb.
38x8.25	8	7 in.	10 1/16 in.	38.73 in.	8.44 in.	17.89 in.	2500 lb. at 50 lb.
38x9.00	8	7 in.	10 1/2 in.	38.18 in.	9.02 in.	17.47 in.	3000 lb. at 50 lb.
40x9.00	8	7 in.	10 1/2 in.	40.18 in.	9.02 in.	18.51 in.	3000 lb. at 50 lb.
Heavy Duty							
30x5	Com'l	5 in.	....	32.25 in.	5.91 in.	15.27 in.	1700 lb. at 80 lb.
30x5		5 in.	6 5/8 in.	32.48 in.	5.92 in.	15.35 in.	1700 lb. at 80 lb.
34x5		5 in.	6 5/8 in.	37.04 in.	6.16 in.	17.65 in.	1700 lb. at 80 lb.
32x6	10	6 in.	7 1/2 in.	34.39 in.	6.81 in.	16.19 in.	2200 lb. at 90 lb.
36x6		6 in.	7 1/2 in.	38.41 in.	6.86 in.	18.33 in.	2200 lb. at 90 lb.
34x7	12	7 in.	8 3/8 in.	36.05 in.	8.05 in.	17.01 in.	3000 lb. at 100 lb.
38x7		7 in.	8 3/8 in.	39.81 in.	7.80 in.	18.96 in.	3000 lb. at 100 lb.
36x8	14	8 in.	10 1/16 in.	38.36 in.	9.17 in.	18.05 in.	4000 lb. at 110 lb.
40x8		8 in.	10 1/16 in.	41.79 in.	8.80 in.	19.85 in.	4000 lb. at 110 lb.

It will be noted that the rim base sizes for balloon or low pressure tires in this tabulation are 20 and 22 in. Also this manufacturer retains a 6 in. rim for the 7.50 in. tire section, whereas many manufacturers have retained the 6 in. rim only for the 6.75 in. section and use 7 in. rims for all the larger sizes.

An additional sidelight covering the relationship of inflation pressure to carrying capacity for the various tire sections is shown by the following tabulation, which also is a tentative standard of the Tire and Rim Association, Inc.

TABLE 2

Pressure lb. 1 sq. in.	Carrying capacity per tire of nominal dimension			
	6.75 in.	7.50 in.	8.25 in.	9.00 in.
40	1500 lb.	1875 lb.	2225 lb.	2700 lb.
45	1700 lb.	2100 lb.	2500 lb.	3000 lb.

Spacing distance when used in dual mounting is subject to a tolerance of 1/8 in. Apparently the Tire and Rim Association is putting its best foot forward by the establishment of another tentative standard which is designed to make nominal sizes mean something other than a purchasing department figure. The following tabulation shows the standards of maximum or limit size which have been adopted. Whether or not they will be respected remains to be seen. If current passenger car practice is any criterion, their value will be largely historical.

TABLE 3

Nominal Tire Size	Limit of Size
6.75	7.05 on 6 in. rim.
7.50	7.85 on 7 in. rim.
8.25	8.45 on 7 in. rim.
9.00	9.05 on 7 in. rim.

While manufacturers are in close accord on the balloon tire sections shown in the previous tables, the rim or base diameter is the subject of much thought and considerable disagreement. Every tire manufacturer sees the desirability of the 20 in. diameter base and all of them are prepared to supply this size but on the other hand there is a pronounced sentiment toward 21 in. and even 22 in. base sizes. The smaller diameter fits excellently into the current trend toward low center of gravity and low floor height but the heat nightmare encountered in high pressure pneumatic operation is responsible for the leaning toward 21 and 22 in. diameters.

Particularly with dual tires, the heat of the brakes has caused countless failures of the inside tires at mere fractions of the normal mileage expectancy. In most designs the brake drum is directly inside the rear tire and rim. The wheel house serves as an enclosure for nearly half the tire. Many metal to metal brakes are used and with high pressure tires must be applied countless times in a single trip to maintain schedules. The average run is a constant alternation of braking and acceleration.

In addition to bitter experience, tests have demonstrated that brake drum temperatures of 550-600 deg. Fahr. are common. While tires are bound to gain in temperature due to the heat of the road and internal flexure, these and similar factors are ridiculously small as compared to the heat transmitted from the brake drums. Tire, tube and flap design and structure comprehend the absorption and radiation of the heat which is proper to their operation but they cannot be endowed with miraculous properties. Without the influence of the brake drum, operating temperatures of 150-160 deg. may be reached but without consequent trouble. The casings and tubes will withstand such temperatures.

Tests and experience have shown that the beads of the inside tire will reach temperatures as high as 300 deg., which is higher than the vulcanization point. Under the same conditions, the temperature at the beads of the outer tire rarely will exceed 120 deg.

What is the solution of this combination of unpleasant riding and almost prohibitive tire expense? The tire maker presents the balloon tire as a definite help but not a panacea. The automotive industry through the designer must re-

Fig. 1. *High and low pressure heavy duty tires of comparable load characteristics. Each mounts on a 20 in. base. At left, 36 x 8.25 balloon and at right, 32 x 6 high pressure tire*

align some ideas and, lastly, the operator must acquire better "tire management."

In the first place, the comfort provisions of the balloon tire will reduce sharply the amount of braking required over a run.

Then the balloon tire has a larger radiating surface and will develop less internal heat due to the thinner walls and

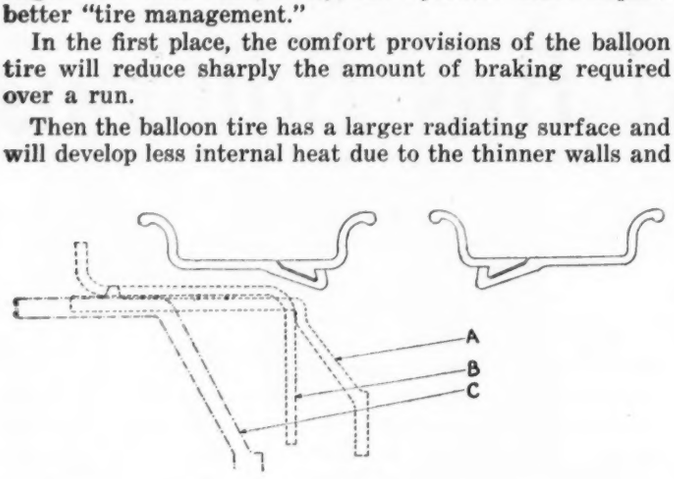


Fig. 2. *Firestone Type-K rim and felloe construction for dual mounting. Brake drum C is a recommended mounting for the elimination of heating trouble at tires*

greater flexibility. Molded rubber flaps are on the increase. All of these will take care of the heat developed within the tire. At this point the work of the automotive engineer begins.

While a material reduction in bead temperature is made by the increase from 20 in. to 21 in. diameter and a further slight reduction can be made with the 22 in. rim, the tendency of the automotive designer will probably be toward the smallest diameter rim. With balloon tires, this tendency will be almost necessary in many cases. Fig. 1 shows a 36 x 8.25 tire at the left which supplants a 32 x 6

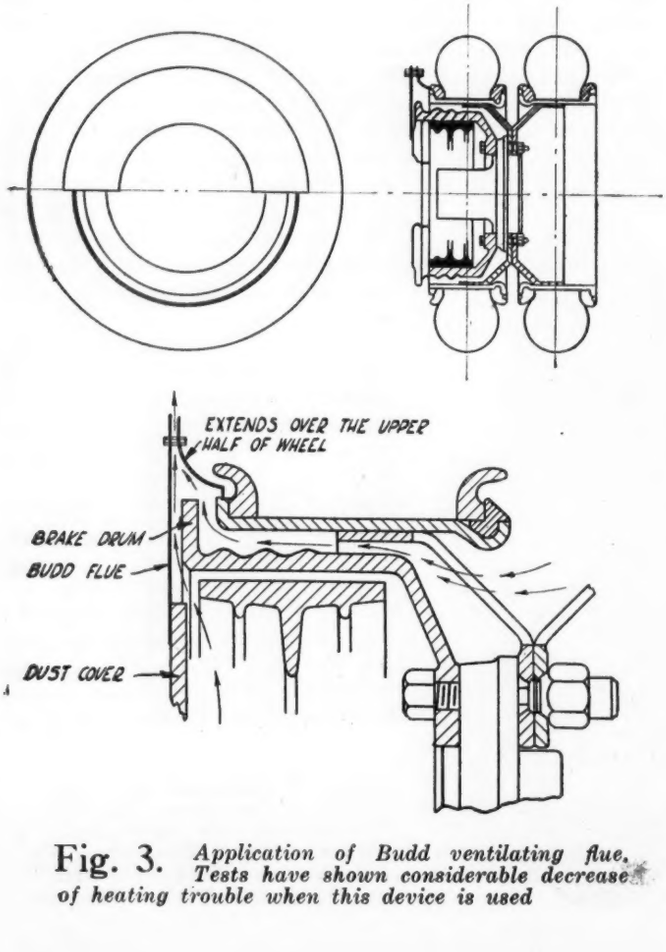


Fig. 3. *Application of Budd ventilating flue. Tests have shown considerable decrease of heating trouble when this device is used*

high pressure tire shown at the right. The problems of wheel housings and inside clearance are self-evident.

Nevertheless the alternatives for the automotive engineer seem to be: Increase the diameter of the rim or reduce the diameter of the present conventional brake. At least ¾ in. clearance should be allowed between the outside of the brake drum and the inside of the rim. In many existing designs having 20 in. rims this clearance is only ¼ in. The answer is tire trouble without end.

As another alternative for the ones mentioned in the previous paragraph, the brake drum can be offset or shifted toward the center with relation to the rim so that the greater portion of its outer periphery is clear of the rim. Designs of this type have been particularly free from tire heating troubles. In Fig. 2, which illustrates the new Firestone type K rim, an excellent example of this construction is shown by brake drum C. In many existing jobs, the same result may be accomplished by spacers or

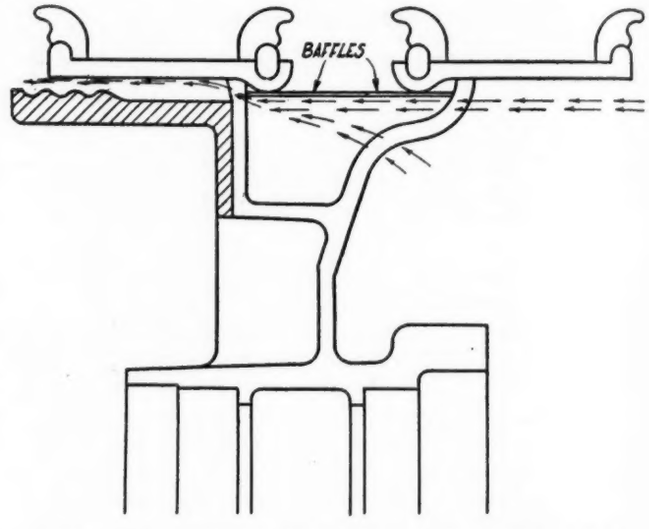


Fig. 4. *Parker wheel equipped with baffles which direct air current through space between brake drum and inner rim*

some other method of shifting the rim outward with relation to the drum.

Tire makers are advocating more general adoption of four-wheel brakes with the idea that brake pressures and therefore brake temperatures will be reduced by splitting up the work of deceleration. Another suggestion is the use of insulating rings of asbestos or similar material around the outside of the brake drums. This solution is questionable from the standpoint of the chassis engineer. Still another suggestion is the use of bronze shoes in metal to metal brakes. Both of these suggestions should be the subject of cooperative effort between the two industries.

In a sense, the problem of eliminating brake drum heat from the rear tires, particularly in dual installations, may be compared to those of bus ventilation and heating. Every available factor must be utilized to insure adequate results.

Two recent developments in the wheel and rim field offer possibilities. Fig. 3 illustrates the Budd ventilating flue developed by the well-known manufacturer of disk wheels and all-steel bodies. Two sheet steel members formed as shown are attached to the upper portion of the dust cover so that the partial circumferential outlet lays just beside the inside edge of the rim. The draft which is induced reduces the tire bead temperature materially.

Another development has been made in conjunction with the Parker cast wheel. As shown by Fig 4, a circular sheet steel baffle is placed between the rims to deflect

the air currents which ordinarily would pass through at this point to the inner edge of the rim and thus pass them over the outside of the brake drums.

One company, Goodrich, has introduced a line of tires with low pressure characteristics applicable to the 24 in. base which are in common use with existing high pressure tires. As the name of the new line, Ovalcord, indicates, these tires are designed to fit into existing diametral conditions but have greater volume and fewer plies.

After the tire engineer has developed the balloon tire and has specified the correct section for a given job and load and the automotive engineer has rounded out a chassis in which external heating influences are eliminated, the element of tire management enters. And with this factor, the problem of education of the fleet owner and driver confronts both industries. Where a tire company

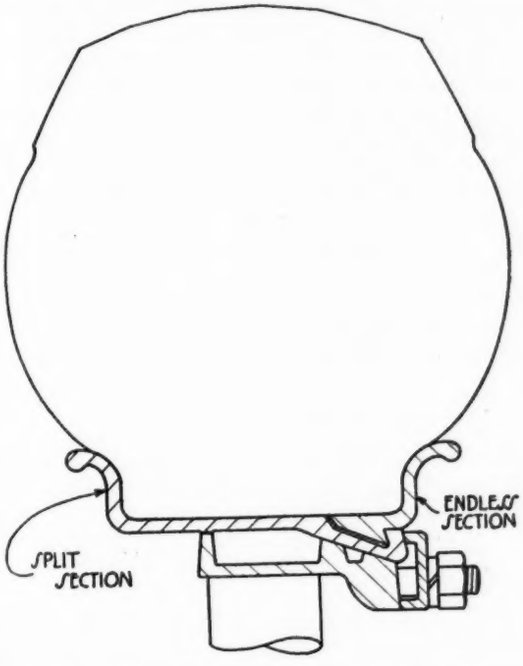


Fig. 5. *Firestone Type-K rim in single mounting. Joint arrangement and locking device simplify tire changes. This type rim opens up new possibilities of bus wheel construction*

or an unusual operator has given the tires intelligent handling, the results have been astounding. Mileage has been increased 60 per cent and more by maintaining correct inflation pressures, wheel alignment and toe-in.

One of the most recent developments is the Firestone type K rim which opens up a field of wide possibilities for wood, cast and fabricated wheels. Unsprung weight is reduced, the mounting of the tire for either single or dual installations is simplified and the labor of tire changing is reduced to a marked degree. As shown in Fig. 2 as a dual installation and in Fig. 5 in a single installation, a rolled rim which comprehends the flat tire base and one straight side flange is split diagonally at a bevel joint. A lip at the opposite edge of the rim has a reverse taper which locks into an endless lock ring. When used in either single or dual installations each rim has a single bevel mounting. The split rim is secured between the bevel on the felloe and the similar bevel on the lock ring.

In dual installations, the inner rim bears on the beveled felloe as shown and a bevel ring drawn up at integral lugs. An endless rolled steel spacer separates the two rims. Driving lugs are formed from the split rim, eliminating welding. Channels are formed in the felloe and valve stem extensions are used to bring both valve caps out through holes placed at the middle of felloe and spacer.

The method of clamping single installations on the felloe is shown by Fig. 5. The various parts are designed to allow interchangeability of 5 in. to 8 in. rims on the same felloe with correct center spacing. A heavy screw driver is the only tool required for removing the tire from the rim. This is used to contract one lip of the split rim. Tire jacks and clamps are unnecessary. In the design of the entire assembly particular attention was paid to clearing existing sizes of bus brake drums. Three existing brake drums are shown in Fig. 2.

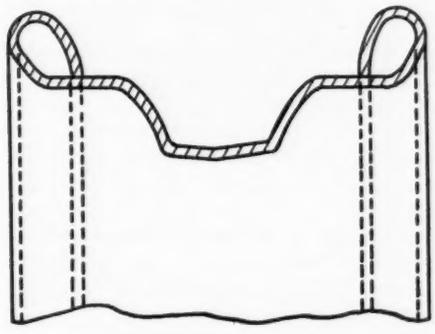


Fig. 6. *Latest Ford type drop center rim construction which reduces chances of incorrect tire location*

Tire makers are not at all excited about the drop center rim and do not anticipate any early broad adoption of this device. While the drop center rim works out excellently with the Ford type of welded wire wheel and produces a rim of light weight combined with ease of tire replacement, the problem of retaining both of these qualities with the artillery and conventional design wire wheel is not simple. Some of the disk wheel makers in this country have developed drop center rims which retain both qualities.

Patents on the drop center rim expired several years ago. This fact may influence some manufacturers toward its adoption. Also the drop center rim supplies a convenient solution of the export tire problem. With high pressure tires Europe never really gave up allegiance to the clincher type for the straight side design which was adopted generally here. Then when the ballon tire came into prominence, Dunlop, the foremost British tire maker, introduced the drop center rim in conjunction with it and has succeeded in popularizing the combination.

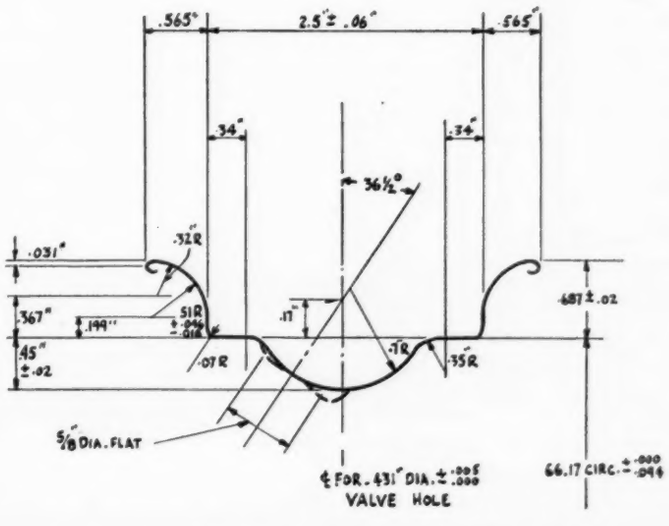


Fig. 7. *New drop center rim section used by Dunlop of England. Well is relatively shallow and designed to insure correct tire location*

From the driver's angle the innovation has been a success. In fact, Maj. Seagrave's Sunbeam special, which recently made a record of more than 152 m.p.h., was equipped with this type of rim. No pronounced difficulty due to running off after a puncture or blow-out has been experienced. Tire changing does not involve rim jacks or clamps. In many cases a new tube is inserted in a punctured casing at the roadside. It will be noted that the Ford spare tire carrier is placed advantageously for this work. As no loose flap is used, the work of changing is simplified and there is no danger of the flap pinching the tube.

New Ford Rim Design

With the earlier Ford drop center rim design there was some danger of not getting the tire properly centered on the rim and casings were made with a red line paralleling the bead base. The owner had to set the tire so that this red line showed above the edge of the rim all around. Otherwise a pinch developed. However, the new design, as illustrated by Fig. 6, eliminated practically all of this difficulty due to the ⅞ in. radius between the sides of the well and the rim base proper. As inflation of the tube proceeds, the beads are forced into proper relationship with the rim flanges.

Recently Dunlop of England has introduced a new drop center rim section which carries the same idea somewhat further. This section is shown in Fig. 7. Narrow seats for the tire beads are adjacent to each side flange and the well is a single large radius which is blended into the bead seats by radii of .35 in.

While many of the American tire manufacturers were forced to decrease the width of their beads in order to make operation with drop center rims possible, their major problem was the development of a tube which would work successively with both drop center and flat bases. Practically every standard tire manufacturer has accomplished this result by shortening the mean circumference of the tube so that it inflates on the drop center base and swells out into the casing. In this way, wrinkles and pinches have been avoided. Where tubes designed for flat base operation were stretched 20 per cent during inflation, the new composite or universal tube stretches 27 per cent. The most difficult portion of the problem was the design of the splice in the tube. Ordinarily this junction tended to stretch less than the balance of the tube. With the new tube an action of this sort would cause buckles and pinches at each side of the joint. However, the tire makers have reached a satisfactory solution of this problem.

Practically no new developments have occurred in the passenger car balloon tire field within the past two years.

No article on developments in the tire engineering field would be complete without mention of the sad story of tire standards. At present there is no such condition. Tire men claim they are at a standstill, that every time one of them talks to the car manufacturer in that direction some other firm gets the business. Therefore rubber companies are building whatever the car manufacturers demand and holding briefs only for the section size for a given weight.

The variation in tire sizes in the passenger car industry is very great.

The estimate of the total number now in use stands at 36. The list is a glowing tribute to the diversity and individuality of human opinion. Some of the high-powered automotive engineers should hear the tire men giggle over their meticulous choice of rim widths.

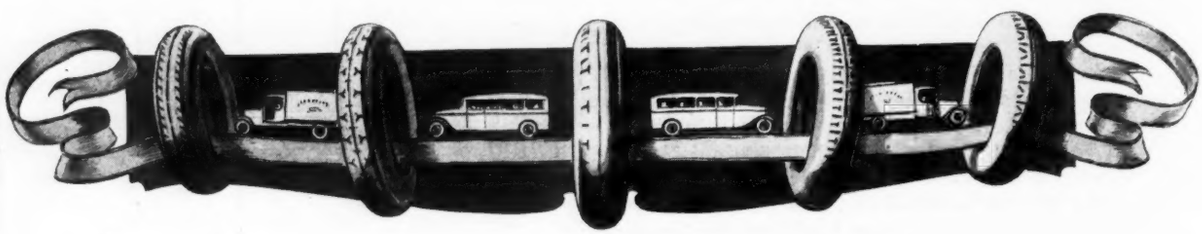
One of the most recent additions is a tire of 27 in. diameter x 4.40 in. section on a rim of 19 in. diameter and nominal 3.5 in. width. The car equipped with these tires is a product of one of the large manufacturers and will make its appearance in the near future. Already tire manufacturers note a strong trend of interest in this small size with inquiries coming in from many producers.

In conjunction with the tendency in car design toward lower center of gravity, smaller engines of higher speed and more sporty appearance, rim diameters are decreasing constantly. When only high pressure tires were available, the prevalent rim diameter ranged from 23 in. to 25 in. With balloon or low pressure tires, 23 and even 22 in. diameters are exceptional and the 20 and 21 in. sizes are about evenly divided, although if the list is weighted by the production of various makers the smaller figure is predominant. Some of the large tire producers anticipate that the 19 in. size will predominate in the not far distant future.

Sizes are Suggested

In conjunction with the remote possibility of standardization effort, one tire development engineer has suggested the following sizes, which cover the requirements of the variety of existing sizes: 4.40, 5.00, 5.50, 6.00 and 6.75. His suggestion covers only the nominal section size. The rim size, of course, will have to vary somewhat, say 19 in. to 21 in., but the solution of rim widths, and in fact the whole situation, is very largely up to the automotive engineers.

During the past year or so, the solid tire of the cushion type or an alternative cushion filler for pneumatic casings when used on light, relatively slow speed commercial vehicles, have gained considerable prominence. Goodrich now is producing the Rubber Ace, which is a two-piece filler for pneumatic casings. Goodyear has just introduced a 30 x 5 demountable cushion tire for commercial vehicles in the Ford, Chevrolet and similar classes. Several of the smaller companies are specializing in the cushion field.



Reducing Compression Pressure of

Diesel Automotive Engines

Will Permit Lighter Construction

Experience has shown that engines of this type perform satisfactorily with 350-375 lb. compression. Materials used must have great strength with a minimum of weight.

By P. M. Heldt

THE possibility of employing Diesel or heavy oil engines for automotive purposes was given consideration many years ago, but it appears that no serious attempts to adapt them for this class of work were made until after the war, the intensive development of engines of this type for submarines and chasers during the war period having paved the way and brightened the outlook for effort along this line.

What is regarded as the first high speed, high compression oil engine was built in 1906 by Diesel & Co., but it was a stationary type. A high speed, low weight engine was built in Russia in 1909 and was intended as a powerplant for the yacht of the European industrial magnate Nobel. It was an eight-cylinder V engine delivering 200 hp. at 600 r.p.m. and weighing about 22 lb. p. hp.

Some work has been done even on engines for aircraft, notably by Junkers in Germany and Attendu in Canada, and if there is any possibility at all of reducing the weight of the engine to a point where it becomes practical for this class of work, there ought to be comparatively little difficulty in meeting the requirements of the heavier class of automotive service, so far as lightness is concerned.

The chief advantages which the Diesel engine holds out are that it is capable of using much cheaper fuels than gasoline, and that it uses a smaller quantity of such fuels per horsepower-hour. In describing the qualities of these engines, it is sometimes asserted that they will burn any fuel that will run through a pipe. The fuel most generally used in this country is a petroleum product, of the general class known as fuel oils.

Compression Can be Lowered

Large Diesel engines for stationary and marine service are designed to operate at compressions of 500-550 lb. p. sq. in., which corresponds to a compression temperature of upward of 1000 deg. Fahr., but by limiting the engine to petroleum fuel and providing an auxiliary ignition apparatus for starting purposes, it has been found possible in automotive type Diesel engines to get along with 350-375 lb. compression pressure. In one respect a high compression pressure is desirable, for, the same as with carburetor type engines, the thermal efficiency of the Diesel increases with the compression ratio, though in the latter it depends also on the "point of cut-off." On the other hand, high compressions are objectionable in automotive type engines because they necessitate making all parts of the engine heavier, and they also make it more difficult to crank the engine in starting.

Thermal Efficiency Higher

The thermal efficiency in any heat engine depends upon the expansion ratio, but in a Diesel engine the expansion ratio is not the same as the compression ratio. In the true Diesel with air injection of the fuel the pressure is generally considered to remain constant as long as fuel is being supplied. This means that the diagram comprises a straight line extending horizontally from the point of maximum compression for a distance equal to the period of fuel injection. If all of the fuel were supplied and burned instantly at the beginning of the power stroke, the cylinder pressure would rise far above the compression pressure. Therefore, in the Diesel diagram a triangular portion is cut off, as it were, at the top end, whereby the area of the diagram representing useful work is reduced, and the efficiency of the cycle therefore is decreased as compared with a theoretical carburetor engine having the same compression ratio. However, this loss is slight as compared with the gain due to the much higher compression ratio (from 12 to 14, instead of about 4.5) so that the actual thermal efficiency of the Diesel is materially higher than that of the carburetor engine.

In standard automotive engines we seldom get a better fuel economy than one hp. per 0.6 lb. of fuel consumed, whereas with moderate-sized Diesel engines, the fuel consumption under the most favorable conditions is below 0.4 lb. p. hp.-h. Both of these figures refer to full load conditions. With the carburetor engine the fuel economy usually drops off quickly as the load is decreased, and though in the Diesel the efficiency also decreases with decrease in load, the rate of decrease is slower, owing to the fact that the same compression is maintained and the expansion ratio actually increases because of the earlier cut-off. For this reason the fuel consumption on the basis of indicated horsepower-hours

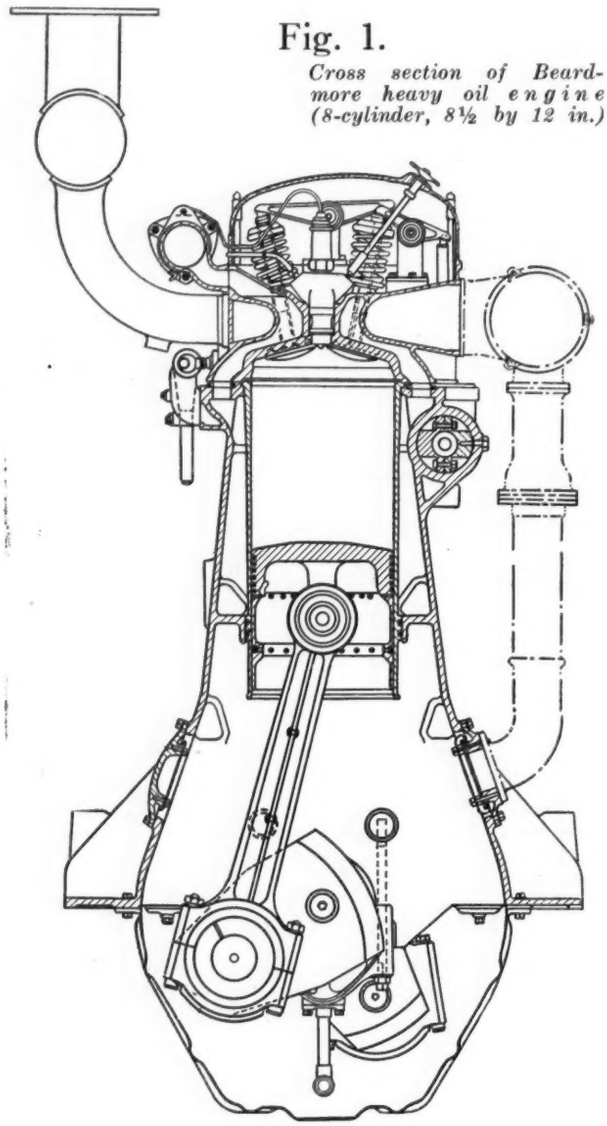


Fig. 1.
Cross section of Beardmore heavy oil engine (8-cylinder, 8½ by 12 in.)

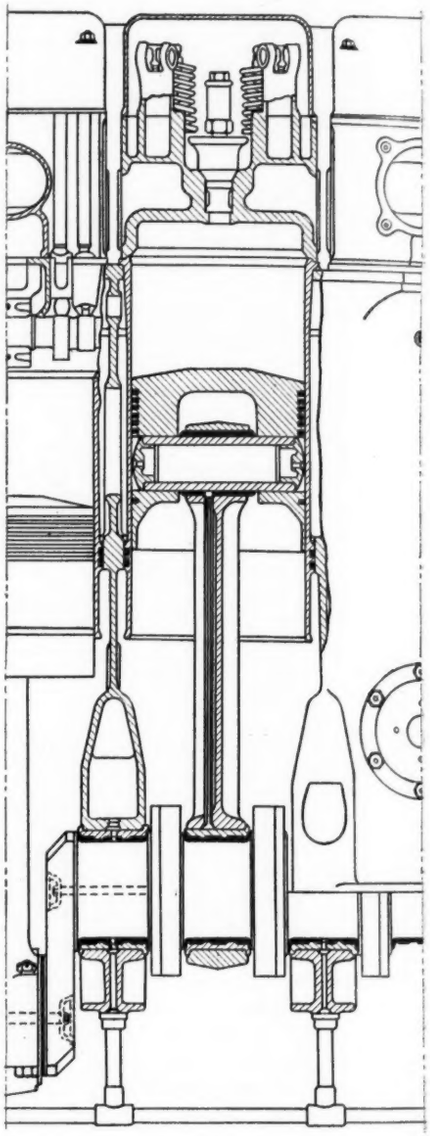


Fig. 2.
Longitudinal section through a portion of the Beardmore engine

decreases as the load decreases, but owing to the power consumed in overcoming internal friction and in driving auxiliaries, which is more or less independent of the load carried by the engine and in many cases quite a large item, the specific consumption on the basis of brake horsepowers naturally increases with a decrease in load.

The first question that comes up when considering a Diesel engine for automotive work is why it should be necessary to build these engines as heavy as stationary and marine engine practice would seem to dictate. With air injection the maximum pressure occurring in the cylinder is of the order of 500-550 lb. p. sq. in., which is very little more than the explosion pressure in a carburetor type of aircraft engine, for instance. That the limiting stresses on the parts in a Diesel engine are far greater than in a carburetor engine can hardly be questioned, and the explanation seems to be that there is a slight chance of the fuel getting into the cylinder too early during the cycle. Since the compression is from four to six times as high, if enough fuel should accidentally get into the cylinder at or before the beginning of the expansion stroke to require all of the air therein for its complete combustion, then an explosion pressure of four to six times that obtained in the carburetor engine would be generated, and the stresses on all parts subjected to this pressure would be exceedingly high.

It is for this reason that in designing stationary and marine Diesel engines it has been customary to provide against stresses due to a maximum gas pressure of 2000 lb. p. sq. in., whereas the normal pressure during injection is only about 500 lb. p. sq. in. and the mean effective pressure only about 100 lb. p. sq. in.

Stronger Construction Necessary

To take care of these high possible or accidental pressures, the combustion chamber walls, pistons and connecting rods, and the crank chamber must all be built much stronger than in a carburetor type engine. Still, for automotive purposes it is essential that the weight be kept down. Reducing the compression pressure from 500 to 375 lb. p. sq. in. helps, for the maximum pressure that must be provided against decreases in direct proportion with the compression pressure. In addition, however, the structural design must be such that the greatest possible strength is obtained with a minimum weight of material.

It will be remembered that the cylinder blocks, heads and crankcases of the Diesel engines on the German submarines, of which two examples were shown to members of the S. A. E. at the Philadelphia Navy Yard in January,

1920, were made of cast steel, and the quality of the castings was generally commented upon. Cast steel is also used for the cylinder blocks of the high speed engine built by Wm. Beardmore & Co., some of which are used in railcars of the Canadian National Railways. In the Beardmore engine four of the cylinders and one-half of the upper part of the crankcase are made in a single casting, and two such blocks, forming the front and rear halves of the engine, respectively, are bolted together in a vertical plane. The cylinder barrels or liners are removable and are of what is described as hard steel, which presumably means steel with a medium carbon content and heat treated before the final finishing, so as to remove all molecular stress and prevent distortion in service. A cross section of the Beardmore engine and a longitudinal section through one cylinder are shown in Figs. 1 and 2 herewith.

Double Cylinder Walls

In the Benz tractor engine, described in these columns some time ago, the structural part (Fig. 3) is made of an aluminum casting into which cast iron liners are pressed. In this case the cylinder walls are double—cast iron surrounded by aluminum—while in the case of the Beardmore the liner is supported only at the top and near the bottom and is in contact with the cooling water over the greater part of its length. In the Maybach six-cylinder engine, which has been used on railcars, the crankcase is carried high up over the cylinder block, or, in other words, the block is sunk into the crankcase, as in the automobile engine of the same make, but since the crankcase is made of cast iron there can hardly be any appreciable saving in weight, and the chief objects of the design seem to be to secure a very rigid structure and a smooth exterior.

In the Attendu aircraft engine both the cylinders and the cylinder heads are aluminum castings, the former being provided with a liner of mild steel and the latter with inserted (not cast in) valve seats of the same material. This being a two-stroke engine, the two valves in the cylinder head both serve as inlet valves, and overheating of the valve seats therefore is not to be feared.

A few of the automotive type engines have a plain cast iron cylinder block, which construction, of course, commends itself from the point of view of low production cost, and only the crankcase is cast of aluminum. The M. A. N. truck engine is of this type.

The use of medium carbon steel for the liners is advantageous because of its greater strength and also because of its better bearing qualities as compared with low carbon steel. For best utilization of the material the wall thickness is gradually increased toward the flanged top end, following the practice that has grown up in connection with steel cylinders of aircraft engines such as the Liberty, of thickening the cylinder wall at the compression chamber.

L-head cylinders are not used for Diesel engines, because with valves in a side pocket it would be difficult to keep the volume of the compression space within the required limit. In the standard Diesel construction the air valve and exhaust valve are both located in the cylinder head, with the fuel injection valve between them. One thing in favor of this arrangement is that the whole of the valve mechanism is arranged together on the cylinder head, leaving the sides of the engine free and thus making for a neat and compact design. In high speed type engines it is difficult, however, to provide sufficient room for adequate valves if the injection valve has to be placed between the air and exhaust valves. One alternative, found in the Beardmore engine, is to use double air and exhaust valves, that is, four valves per cylinder, which naturally leaves room at the center of the cylinder head for the injection valve. Another way out is to fit the injection valve or valves into the side wall of the cylinder.

Injection of the fuel into the combustion chamber against the high pressure of compression may be effected by air pressure, by partial combustion of the fuel charge

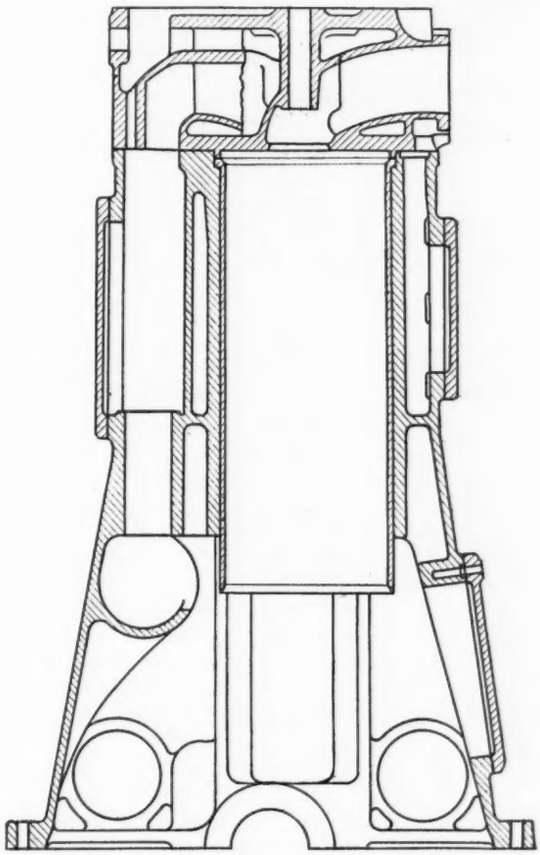


Fig. 3. *Section through cylinder block and liner of Benz engine*

in an ante-chamber, or directly by a pump. All of the earlier Diesel engines had air injection. So far as the character of the resulting combustion is concerned, air injection has proved most effective, but it involves serious complication as well as a material loss in power, particularly when running at small load. The injection air in standard Diesel engines is compressed to from 1000 to 1200 lb. p. sq. in., and this compression is accomplished in two, three or four stages. Moreover, an air reservoir must be used in which the compressed air is stored, and it is also necessary to provide inter-coolers and an after-cooler, as the air is heated during each stage of compression and must be cooled before it enters the next stage or the reservoir in order to increase the efficiency of the compression process and prevent explosions due to the absorption of lubricating oil. In fact, to guard, against such explosions, an oil extractor is usually provided.

Air Injection Simplified

Where air injection has been used in connection with small engines the process generally has been simplified to a considerable extent. For instance, in the Hindlmeyer engine (Fig. 4) developed in Austria, the injection air is compressed to 700 lb. p. sq. in. pressure in a single stage and is used directly without being stored.

For multi-cylinder engines, which alone can be considered for automotive purposes, this involves the use of a separator compressor for each cylinder. Perhaps the best known of the automotive engines with air injection is the Maybach, which has been used for railcars.

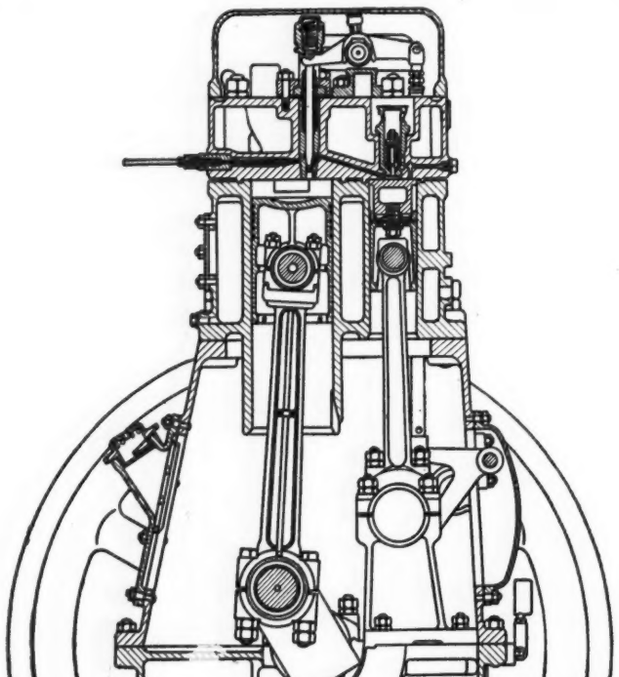


Fig. 4. *Hindlemeyer engine with air injection*

This is a six-cylinder engine developing a maximum of 150 hp. at 1300 r.p.m. and weighing about 2640 lb. or 17.6 lb. p. hp.

Maybach Air Compressor

The air compressor of the Maybach engine is of the three-stage type and is arranged at the forward end of the engine, its crankshaft being coupled to the engine crankshaft. This compressor cools the air after each stage of compression. In the admission line there is located an automatic throttle valve which keeps the pressure of the air delivered constant under all conditions. The valves of the compressor are said to be of such design as to prevent interference with their operation by carbon particles, etc. With the engine itself, the compressor shares the features of a completely enclosed design and the use of roller bearings on the crankshaft. This engine, when operated under conditions of maximum efficiency, that is, under normal load at 800 r.p.m., consumes about 0.375 lb. of fuel per hp.-h. The specific fuel consumption increases with any change in load either up or down, and also with any change in speed, though the effect of moderate changes in speed is not very marked.

Mechanical injection, or, as it is often called, solid injection, which was originally introduced by Vickers in England, permits of a considerable simplification of the engine. In the first applications of this method of injection the engine had the conventional flat cylindrical compression space, with the injector valve mounted centrally in the cylinder head. It was found that with this arrangement the atomization and combustion of the fuel were quite faulty, and it was said at that time that a motor ship with solid injection could be immediately picked out at sea by the smoky exhaust emitted from its funnel. Remembering that the compression ratio is nearly 15 to 1, if the stroke bore ratio is 1.5, the compression space is substantially a cylinder of a depth equal to one-tenth its diameter. If the fuel is injected into such a space along the cylinder axis, it almost immediately strikes the piston head, and this appears to have a chilling effect which condenses the fine spray and causes it either to combine into larger globules or to adhere to the metal, thereby preventing its uniform distribution througout the mass of the air, and its complete combustion. The smoky exhaust, of course, is indicative of incomplete combustion; with air injection the exhaust from a Diesel engine is normally absolutely invisible.

Atomization of Fuel

The statement that air injection results in better atomization and more complete combustion than solid injection was meant to convey that this has been generally the case in the past and not that this is necessarily and unalterably so. Atomization and distribution of the fuel are affected by the form of the combustion chamber, the number and location of the injectors, the form of the injector outlets, the pressure behind the jet, etc. Numerous modifications are therefore possible, and much of the work which has been done on oil engines in recent years has centered around the parts which affect the atomization and distribution of fuel with solid injection.

In England, where solid injection originated, the flat combustion chamber originally used was replaced by a substantially hemispherical one of the general form shown in Fig. 5. At the same time the injector valve was provided with a number of openings inclined to the axis of the cylinder, and the pressure behind the jet was so adjusted that the jet was about spent when it reached the piston wall.

The Krupp Works in Essen have developed a solid injection system in recent years and claim to have found a solution whereby it is posible to completely burn the fuel in a flat cylindrical combustion chamber without any appreciable increase in pressure. They make use of what might be called a flash plate, that is, a mushroom-shaped piece of cast iron secured to the piston head but having only a comparatively small area of contact with the latter, so as to impede the heat flow from the plate to the piston. The central portion of the plate is convex

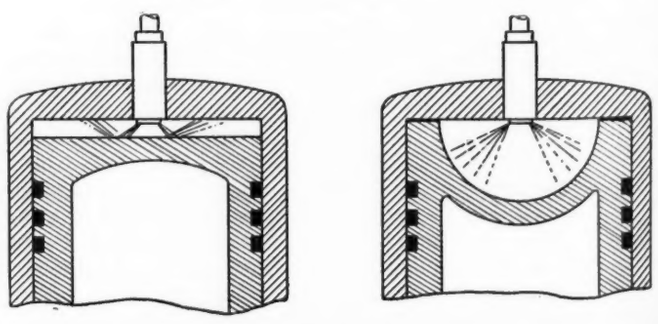


Fig. 5. *Forms of combustion chamber and arrangements of injector nozzles*

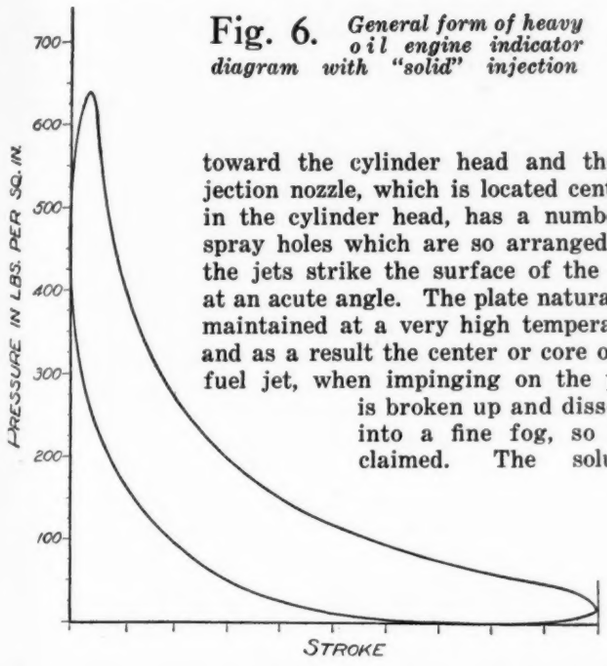


Fig. 6. *General form of heavy oil engine indicator diagram with "solid" injection*

toward the cylinder head and the injection nozzle, which is located centrally in the cylinder head, has a number of spray holes which are so arranged that the jets strike the surface of the plate at an acute angle. The plate naturally is maintained at a very high temperature, and as a result the center or core of the fuel jet, when impinging on the plate, is broken up and dissolved into a fine fog, so it is claimed. The solution, however, can hardly be considered for high speed engines because the flash plate would add too much to the reciprocating masses.

With solid injection or mechanical injection it appears difficult to so regulate the supply of fuel that the pressure within the combustion chamber shall remain constant during the whole of the injection period, and the indicator card from a Diesel engine with solid injection is intermediate in form between an indicator card from a carburetor type engine and that from a constant pressure engine. That is, upon the ignition of the charge there is a material increase in pressure above the maximum compression pressure. Fig. 6 shows a full load indicator diagram from an oil engine with solid injection.

In Germany extensive use has been made of what is referred to as a Vorkammer or ante-chamber, into which the fuel is injected and which communicates with the combustion chamber through one or more fine openings. This method of fuel injection was originally associated with the name Brons, while in this country it has been known under the name of Hvid.

In the Brons and Hvid engines, which work on the four-stroke cycle, during the suction stroke air alone is drawn into the cylinder, but at the same time fuel mixed with a little air is drawn into a cup secured to the cylinder head on the under side and which communicates with the cylinder through a small opening. During the compression stroke the air is compressed to 450 lb. p. sq. in., the compression is communicated to the interior of the cup, and the fuel mixture therein is ignited. The resulting rise in pressure in the cup forces the remaining fuel through the small opening into the cylinder, where it is burned during the following down stroke.

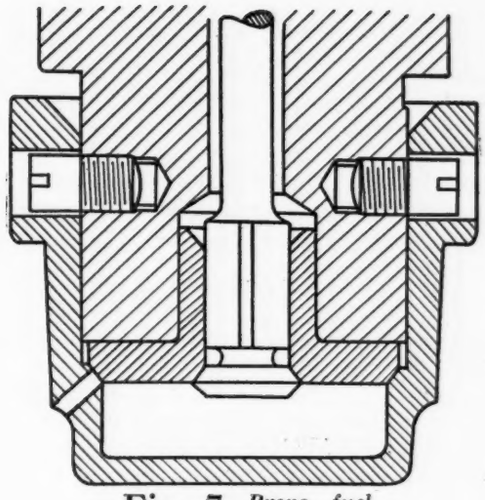


Fig. 7. *Brons fuel cup*

Engines of this type have been built by the Deutz Gas Engine Works for about fifteen years, chiefly for marine work, but the writer has found no record of any high speed engines of this type. Fig. 7 shows the Brons preignition cup. The Hvid engine was introduced in this country about 1920 but has not been developed for automotive purposes.

In later applications of the ante-chamber principle, as in the Benz, Mannheim and Kamper, illustrated in Figs. 8, 9 and 10, fuel is injected into the ante-chamber toward the end of the compression stroke by means of the fuel pump. Between the ante-chamber and the cylinder there is a bored passage into which is inserted a steel thimble which is open toward the ante-chamber and closed toward the cylinder, except for a number of small holes in the head of the thimble. The thimble is made of such outside form that it has comparatively little contact with the cylinder wall, so that the flow of heat from it to the cylinder wall and jacket is impeded, and it, consequently, reaches a high temperature. Aside

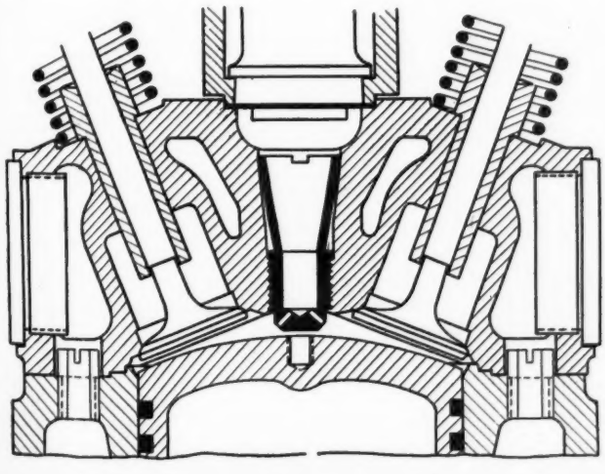


Fig. 8. *Antechamber of Benz engine*

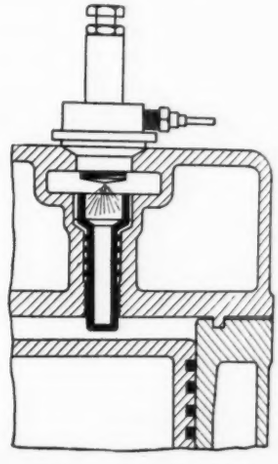


Fig. 9. *Antechamber of Kamper engine*

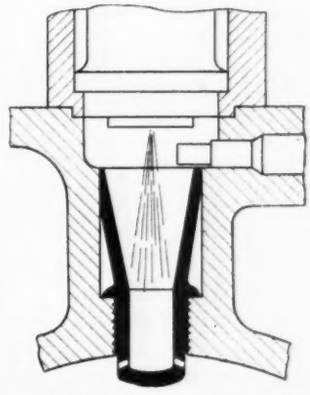


Fig. 10. *Antechamber of Mannheim engine*

from the fact that the fuel is not drawn into the ante-chamber by the suction of the intake stroke, the action is substantially the same as that of the Brons cup. The openings in the thimble are generally located in the plane of the junction between the cylinder and cylinder head and make a considerable angle with the axis of the thimble, hence not only is the fuel injected into the combustion chamber at the very high pressure due to the explosion in the ante-chamber, but it is well distributed.

The Steinbecker Method

Still another variation of the ante-chamber idea is incorporated in the Steinbecker engine which was described in *Automotive Industries* of March 3, 1921. Here the fuel is injected into the throat between the ante-chamber and the combustion chamber. Injection begins toward the end of the compression stroke, and the fuel is carried along by the rush of air through the throat into the ignition chamber where it is ignited by contact with the hot walls. The increase in pressure in the ignition chamber due to combustion then causes the flow through the throat to be reversed, and during the latter part of the injection period the fuel is carried directly into the combustion chamber.

The conditions affecting ignition naturally are most difficult when the engine is being started from cold in cold weather, and if the engine is to start on self-ignition, this is the condition which determines the compression which must be carried, and hence the maximum pressures which the engine has to withstand. To be able to reduce the compression, a number of engines use an auxiliary ignition device for starting Two such ignition systems are in use. One consists of a coil of nickel-chromium wire located in the combustion chamber and provided with an insulated lead passing through the combustion chamber wall, which coil is maintained at a red heat by current from a battery. The other system consists in the use of an ignition plug or cartridge consisting of a wad of paper impregnated with potassium nitrate, which is ignited by a match before it is inserted into the engine. A new plug must be inserted each time the engine is to be started from cold. The arrangement of such an ignition plug is shown in Fig. 10 in the illustration of the pre-ignition chamber of the Mannheim engine.

The fuel feed system of a Diesel engine usually comprises two elements, viz., the pump and the injection valve. A multi-cylinder pump is generally used. With the original Diesel engine with air injection the injector valve is opened positively by means of a cam, but in most present-day engines the valve is opened by the increasing pressure of the fuel due to the pump action, and is closed by a spring. The valve has a conical seat and an enlarged section of the stem, which latter acts as a piston. The spring normally holds the valve against its seat, but when the pressure on the fuel reaches a certain value, its pressure on the piston or flange on the valve stem overcomes the pressure of the spring and the valve opens.

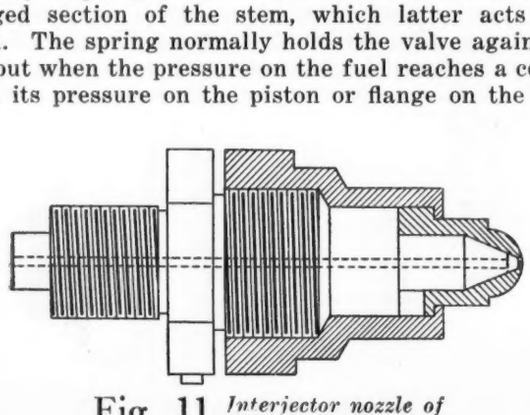


Fig. 11. *Interjector nozzle of M.A.N. engine*

A departure from this practice is made in the M.A.N. four cylinder truck engine, which has two injector nozzles in each cylinder, these taking the place of the usual spark plugs, being of substantially the same size. There is no valve in this nozzle, which communicates directly with the pump outlet through a small bore tube. Thus the delivery valve of the pump is the shut-off valve of the fuel injector, as it were. The M.A.N. injector nozzle is very simple, consisting merely of three parts which are screw machines products, namely, the nozzle shell, the nozzle core and the nozzle cap. A sectional view of the asembly is shown in Fig. 11. The outer end of the core screws into a plate of oval shape which forms a flange for fastening the nozzle to the cylinder block by means of two bolts, into which flange the fuel pipe is secured.

The fuel pump is generally regarded as one of the most difficult design problems in connection with high speed Diesel engines, because of the minute quantities of fuel which it must deliver, in accurately measured doses, accurately timed. With the heavier hydrocarbons the volumetric proportion of liquid fuel to the air required for its complete combustion is about 10,000 to 1, so the maximum amount of fuel which the pump must deliver per cycle is about 1/10,000 of the piston displacement, and for idling and small loads the fuel supply must be further greatly decreased.

The pumps, as a rule, are plain plunger pumps. Many of the pumps have a considerable number of grooves turned on their plungers, which help to prevent leakage. In the Maybach six-cylinder engine, any fuel that leaks by the plungers of the six-cylinder pump collects at the bottom of the pump housing and is returned to the fuel tank by a leakage pump. No packing is used in this pump.

Like other commercial vehicle engines, Diesel automotive engines are generally provided with a governor to prevent them from racing while idling. This governor acts by varying the fuel supply, the amount of air drawn in remaining the same irrespective of the load. The amount of fuel supplied by the pump per stroke can be varied either by varying the pump stroke or by keeping the suction valve of the pump open during a greater or smaller fraction of the delivery stroke. In the past the usual method has been to have the governor act on the pump inlet valve; this is both lighter and smaller than the plunger and its associated parts, and it therefore seems reasonable to assume that the governor can be made more sensitive when acting on the valve than when acting on the pump drive.

Pump Timing Changed

In cases where the fuel pump is operated by a cam, the fuel supply can be varied by some means which prevents the pump plunger from following the cam all the way when on the down slope. A disadvantage of this principle is that the timing of the beginning of the injection period also changes when the cut off is changed. In the Attendu fuel pump this effect is obviated by simultaneously changing the timing of the pump by means of a sliding coupling involving the use of helical splines and a planetary gear arrangement.

Recently a third method has come into favor, involving the use of a flash valve which is moved to suddenly open or block a by-pass or return passage for the fuel from the delivery side of the pump. This principle has been used in Germany by Deutz and M.A.N., who employ a by-pass from the delivery to the inlet or suction side

of the pump in which the flash valve is placed. In England the Beardmore Company uses a similar system which is illustrated in diagram in Fig. 12. Normally the pump plunger, which is actuated by an eccentric, moves the fuel back and forth in the passage from the supply tank to the pump cylinder. At precisely the right moment this pasage is closed by the flash valve, which is of a type that can be brought up to speed before it begins to close the port, so that the closing takes place very rapidly. There is then a sudden increase in the pressure on the fuel, and the injector valve is opened very quickly. A sharp cut-off is said to be effected by a reflex action drawing the injector valve back to its seat.

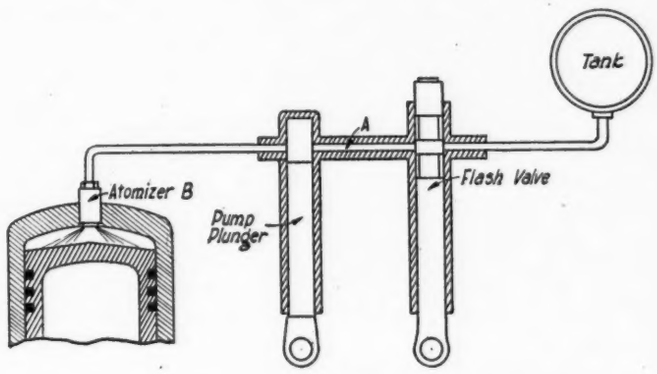


Fig. 12 *Diagram of fuel injection control system used on Beardmore engine*

In the German applications of the principle the flash valve opens the by-pass, thus controlling the cut-off directly. In this way a very sharp cut-off is said to be obtained. This is desirable because any fuel that gets into the cylinder after the completion of the normal period of injection is not utilized to the full advantage. The shorter the period of injection the higher the thermal efficiency, but since the maximum pressure increases as the period of injection is decreased, a compromise is made. With this arrangement the beginning of the period of injection is determined by the pump stroke and is invariable, while the cut off is determined by the cut-off valve in the by-pass and varies according to the engine load.

Heavy Tubes Required

In view of the very high pressures on the fuel with solid injection (pressures of several hundred atmospheres are used by Vickers) it is necessary to make the tubes carrying the fuel to the injector valves with comparatively heavy walls, so that the delivery will not be affected too much by expansion of the tubes.

An interesting combined fuel pump and injection valve due to Dr. Frey is shown in section in Fig. 13. This is used on a two-stroke engine but evidently could be equally well adapted to a four-stroke engine. In the lower part of the figure will be seen a shaft carrying a pump-operating cam which, through the intermediary of a rocker lever, actuates the plunger of the fuel pump. Directly above the plunger is located the suction valve, which is opened mechanically from the pump plunger through a linkage and closed by a spring and the pressure of the fuel during the delivery stroke. The rounded end of the rocker lever can be moved laterally a certain distance, so that its center is no longer directly over the camshaft axis. By making the rising incline of the cam of tangential form, as shown, this does not affect the beginning of the injection period, but it reduces the stroke of the pump plunger (and, consequently, the amount of fuel delivered) and also the point of cut-off. The horizontal needle valve in the upper part of the drawing is the injection valve, which is opened automatically by the pressure of the fuel against the enlarged portion of its stem.

In Diesel engine development during the past decade and a half the two-stroke cycle engine has played a comparatively important part. One of the chief objections to the two-stroke cycle with carburetor type engines is absent in the case of the Diesel. In the former the charge transfer and exhaust ports are open simultaneously, and it is impossible to prevent some of the charge entering through the transfer port from escaping through the exhaust port, thus wasting fuel. In the two-cycle Diesel nothing but pure air enters the cylinder during the inlet period, and it is therefore impossible for fuel to get into the exhaust before it has had a chance to burn. Moreover, it is possible without extra complication to completely clear the cylinder of the dead gases of combustion by air scavenging, thus producing a supercharging effect and adding to the output of the engine. Since there is one power stroke in each cylinder per revolution, instead of one during two revolutions, and more charge is introduced into the cylinder during every power stroke, a two-stroke Diesel engine has a much greater output than a four-stroke of the same displacement and running at the same speed. It is for this reason that the two-stroke Diesel has become quite popular for marine work.

Most of the engineers who have devoted their energies to the development of oil engines for automotive purposes have selected the four-stroke cycle, believing that with it, since the time of a complete piston stroke is available for filling the cylinder with air, higher speeds are possible. Still, a good many of the recent developments have been on the two-stroke principle. The aircraft engine built for the U. S. Navy by Andre C. Attendu is a two-stroke engine; so is the Tartrais developed by the Peugeot firm in France and an engine known as the Maedler which is under development in Cleveland. In

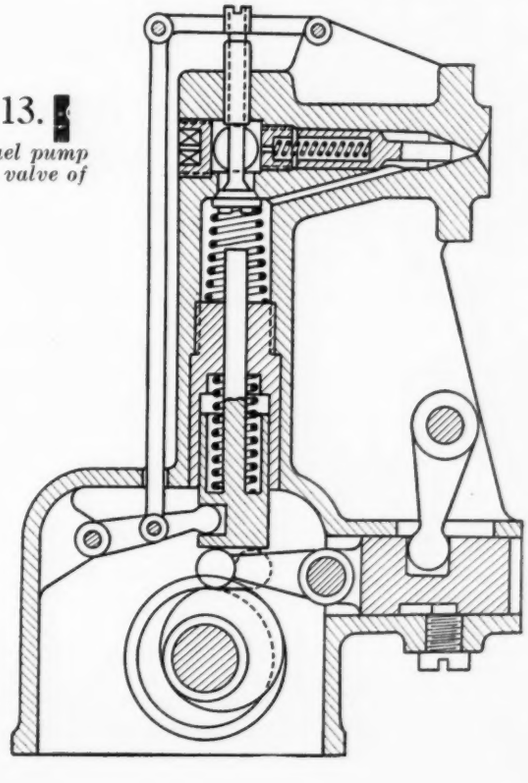


Fig. 13. *Combined fuel pump and injector valve of Dr. Frey*

Germany two-stroke engines of high speed and low weight have been built by at least two parties (Dr. Frey and Hannoversche Waggonfabrik).

When thinking of a two-stroke engine one usually imagines a very simple design without valves and valve gear, but the two-stroke heavy oil engine has little in common with this rudimentary type. In the Attendu engine, for instance, there are two inlet valves and two exhaust valves per cylinder, in addition to piston-controlled exhaust ports. Every two-stroke engine, moreover, must have a scavenging pump. But since the air is forced into the cylinder by a positive displacement pump and there is no posibility of loss of fuel through the intermingling of the fresh charge and the burned gases, the limitations as to speed of the ordinary two-stroke engine seem to be largely eliminated. Scavenging pumps are usually made double-acting, and they require a valve which alternately places the two ends of their cylinders in communication with the inlet manifold.

One feature of the Attendu engine deserves special mention, and that is the variable timing of the exhaust valves, whereby the compression of the air in the cylinder is varied. For various reasons it is desirable to limit the compression pressure in the cylinder under the different conditions of operation to the lowest which will ensure reliable ignition. This calls for a somewhat higher compression in starting and when running under small load, as the cylinder walls are then cold, and the loss to the cylinder walls is increased also on account of the lower speed. Attendu secures this additional compression pressure under the conditions mentioned by closing his exhaust valves slightly earlier, so that the cylinders contain a greater amount of air at the beginning of the compression period. The beginning of the exhaust period is not affected by this action on the exhaust valves, as the latter are always open when the exhaust ports are uncovered by the pistons.

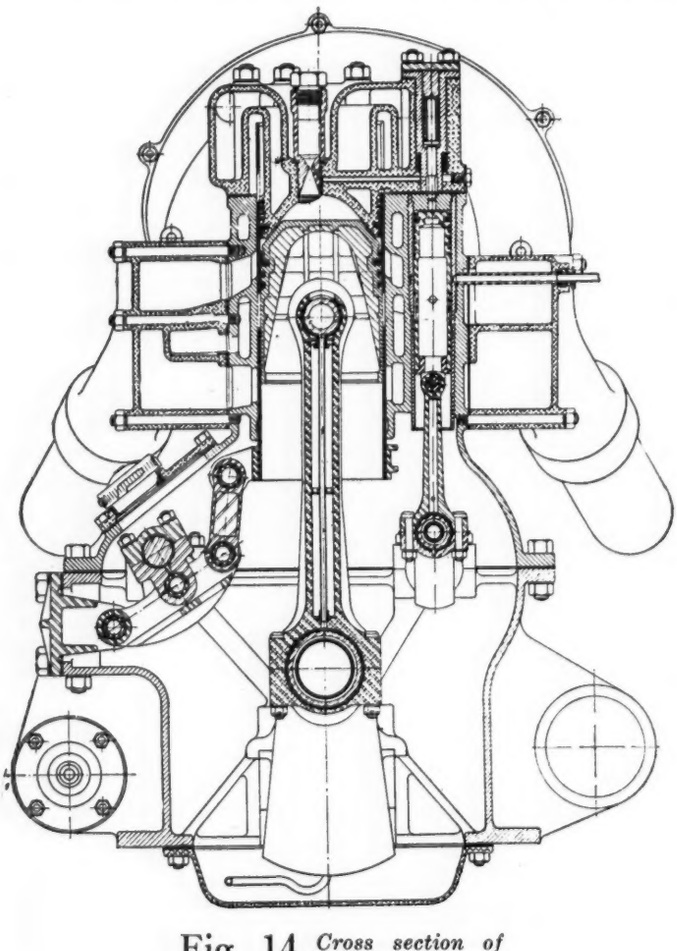


Fig. 14. *Cross section of Maedler engine*

A high pressure oil engine that embodies some rather novel ideas is the Maedler, which is being developed in Cleveland. A single cylinder experimental model of 4¼ in. bore and 5½ in. stroke has been built and a section of this engine is shown in Fig. 14. Parallel to the power cylinder there is a pump cylinder of something like one quarter the bore. The engine works on the two-stroke cycle and the pump plunger makes the same number of strokes per minute as the piston in the power cylinder.

When the pump plunger approaches the end of its down stroke, at which time there is a high degree of vacuum in the pump barrel, it uncovers a port through which inert gases (exhaust gases) can flow into it. These gases draw in with them a certain amount of liquid fuel which is drawn from a spray nozzle as in the conventional carburetor. The vacuum in the pump chamber when its inlet port is uncovered is always the same, and hence the same amount of exhaust gas and the same amount of fuel will be drawn into the pump barrel per engine cycle under all conditions of speed and load.

During the up-stroke of the pump plunger, which coincides with the up-stroke of the power piston, the charge of dead gases and fuel is compressed in the pump barrel to a presure of about 600 lb. p. sq. in. Owing to the heat generated by this compression, the fuel is thoroughly vaporized and mixed with the carrying gases, but it cannot ignite because there is no oxygen present. About 10 deg. before the upper dead center position is reached, the pressure in the pump barrel opens the spring-loaded valve above it, and the fuel charge then flows through a fine passage, formed by a copper-plated steel tube which is set into the casting of the cylinder head to a venturi passage connecting the compression space proper with a so-called ignition chamber. As the fuel mixture begins to issue from the transfer passage, air is being forced from the working cylinder into the ignition chamber and attains a very high speed at the throat of the venturi. After a certain amount of fuel has been discharged, ignition takes place in the ignition chamber spontaneously, and the resulting increase in pressure causes a reversal of flow through the venturi.

In this particular design the ignition chamber has only one sixth the volume of the total compression space, so that only a small portion of the air gets into the ignition chamber. During the latter part of the injection period the fuel mixture, therefore, flows from the throat of the venturi directly into the combustion chamber.

It will be noticed from the cross sectional drawing that the engine is provided with a single sleeve valve. This opens the exhaust ports well ahead of the completion of the expansion stroke, and after the pressure within the cylinder has been reduced to near atmospheric, the air inlet ports are opened. A blower or supercharger is used with the engine, which scavenges the combustion chamber during the first part of the inlet period and then fills it with fresh air to slightly above normal atmospheric pressure, thus affecting a certain degree of supercharging.

During the early part of the down stroke, after the contents of the pump barrel have been delivered to the venturi, the spring-loaded valve closes and communication with the cylinder is thereby interrupted.

The illustration of the Maedler engine here shown is of an older experimental model. The company is now working on a six-cylinder 125 hp. bus engine embodying the same method of fuel injection, and we expect to print a description of this later design in an early issue.

Small, Supercharged Cars for Future Predicted at S. A. E. Gathering

Engineers evince growing interest in technical experiences developed by use of "toy motors" in racing. J. H. Hunt nominated for presidency during 1927.

By Norman G. Shidle

THAT the technical experiences developed by racing are of growing interest to stock vehicle engineers and that headlighting development still is an outstanding automotive engineering problem were evidenced plainly at the Summer Meeting of the Society of Automotive Engineers held at French Lick Springs, Indiana, June 1-4. Over eight hundred members and guests attended the sessions at which other technical questions such as riding qualities, fuels and lubrication, anti-freeze developments, gears and drop center rims for balloon tires were discussed, in addition to the two topics already mentioned.

J. H. Hunt, head electrical division, General Motors Corp. Research Laboratories, was nominated to be the next president of the society. He will move to the presidency from the position of first vice-president, which he now occupies, at the beginning of 1927.

Col. W. G. Wall, consulting engineer, of Indianapolis, was chosen as first vice-president for next year, assuming the office left vacant by Mr. Hunt.

Other officers nominated are: Second vice-president, representing motor car engineering, Fred M. Zeder, vice-president in charge of engineering, Chrysler Corp.; second vice-president, representing tractor engineering, J. F. Max Patitz, chief consulting engineer, Allis-Chalmers, Mfg. Co.; second vice-president, representing aeronautic engineering, E. T. Jones, Wright Aeronautical Corp.; second vice-president, representing marine engineering, P. G. Zimmerman, plant manager, Sea Sled Corp.; second vice-president, representing stationary internal combustion engineering, C. B. Jahnke, chief engineer, Fairbanks Morse & Co.; treasurer, C. B. Whittelsey, president, Hartford Rubber Works; councilors, (to serve for two years) W. S. Sparrow, Bureau of Standards; C. B. Veal, Manly & Veal, and E. Wooler, chief engineer, Timken Roller Bearing Co.

The very real interest being taken by all of the engineers in racing developments was shown by the liveliness of the discussion at the race session in which Fred S. Duesenberg and Harry Miller were the chief speakers, and by the frequency with which the aftermath of the Indianapolis Speedway classic was found to be the center of lobby and lunch table conversations.

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Direct reference to the relation of racing developments to probable future stock car design was made by President T. J. Litle, Jr., in his talk before the general session on the opening night.

"From what I saw at Indianapolis," Mr. Litle said, "I believe that we will soon see on the American market some of these highspeed toy motors as some call them, equipped probably with some form of supercharging device. These will fit very nicely to the cars of the future which must of necessity be lighter and smaller than any we now use, but which will be luxuriously equipped. The super-craftsmanship of America will be reflected in their design.

Next President of S. A. E.

J. H. Hunt

Mr. Hunt, who has been nominated for the presidency of the S.A.E. during 1927, was born in Saranac, Mich., on March 4, 1882. He was graduated from the University of Michigan in 1905 with the degree of bachelor of science in electrical engineering. His first position after graduation was with the Western Electric Co. Later he spent a year as an instructor at Washington University and five years at Ohio State University. He joined the Packard Motor Car Co. in 1912. In 1913 he became research engineer for the Dayton Engineering Laboratories. Since 1920 he has been doing research work for General Motors, being at present head of the electrical division of the General Motors Research Laboratories. He is serving this year as first vice-president of the S.A.E.

"To accomplish the desired result," he went on, "we shall have to use some of our aluminum alloys and light steel parts to replace some of the cast iron parts of low mentality design. Our manufacturers are not particularly active along these lines today. Some of them have been entirely too conservative in perpetuating antiquated designs that are not fundamentally good, changing only when the public ceased to buy their product."

The possibilities of commercial adoption of superchargers in the near future and the chances of adoption of some two-cycle engines on stock vehicles were the two phases of recent racing development which received most attention at the meeting. More than one engineer professed to see a very considerable likelihood that superchargers will be a part of regular commercial passenger car engine design before very long, while the recent work on two-cycle jobs done by Duesenberg seems to have revived a very real interest in this type of power plant, at least on the part of the technical men.

Rear Axle Drives

From the standpoint of practical automotive engineering one of the most timely subjects discussed at the meeting was that of rear axle drives. Some years ago we arrived at a point where 100 per cent of all passenger car models in production in this country had bevel gear drive—either spiral bevel or straight bevel. However, the bevel gear drive and the location of the propeller shaft which it necessitates had become an obstacle toward further progress in the movement toward lower chassis, which was made desirable by the constantly increasing speeds of our passenger cars.

One manufacturer has gone to the worm drive with undermounted worm and many others are said to have experimental cars out with this type of final drive. On the other hand, a type of gear drive intermediate between the spiral bevel gear and the worm drive has been developed and made commercially available. The subject of final drives for passenger cars may therefore be considered a live one, and the character of the discussion showed the interest taken in the subject by passenger car engineers and to a lesser degree by truck engineers.

The great detail with which various aspects of vehicle design and fuel and lubrication problems are being studied was evidenced more strongly than ever by the very comprehensive way in which various questions, relatively small in scope from one point of view, were treated. The growth in volume and value of the data which is coming out of the research laboratories; the increased activity in trying to predetermine causes of troubles likely to arise in operation; and the analysis of test results as a basis for seeking means of trouble prevention—these were the aspects of engineering trends most strongly emphasized by the gathering from a general viewpoint.

Few startling facts were developed in the technical papers; for the most part they dealt with minute details of specific problems. But the whole meeting tended to impress the wealth of actual data which is being unearthed as regards the smallest problems of automotive design and operation and to emphasize the fact the automotive engineer of the future will have far more exact material upon which to base his efforts than ever has been available in the past.

Talking before the general session, C. F. Kettering showed the importance of speculative thinking about future problems, said that research will have a greater part in future activities than in past, and predicted that as long as the American people preserve their healthy desire for things that they haven't got and are willing to work for them we will continue to have prosperity. He pointed

out that men who have made great discoveries in various scientific fields frequently have been those who were not familiar with the so-called laws of the particular science as those laws were understood by the authorities of their time. The man without the professional's training, Mr. Kettering said, sometimes has a better chance of going ahead because he hasn't already been convinced by the so-called laws that a thing cannot be done.

When a man finds that he cannot solve a problem, Mr. Kettering said, the trouble isn't that the problem is too hard but that the man is too soft.

Socially the meeting was successful from every standpoint. Hotel accommodations were entirely adequate and the confusion as regards rooms which existed last year was entirely lacking. Fine weather lasted throughout the meeting except for a little rain which marred the third day's sports slightly. Terrific heat, which many members had feared because of their previous experiences at a West Baden meeting a few years ago, was noticeable by its absence and Dulcy was able to remark with perfect truthfulness that "The days were warm but the nights were cool and we slept under blankets practically every night."

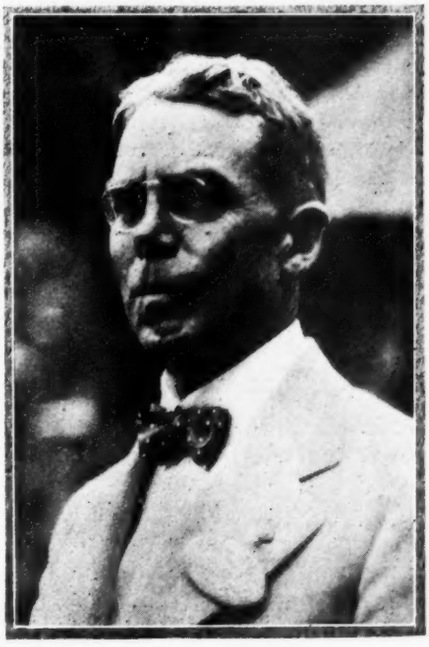
A very large number of full members of the society came to the French Lick sessions. At the time when 783 registrations has been turned in an analysis showed that 314 were full members and 159 associate members. The attendance included five past-presidents of the society.

Golf Most Popular Sport

Golf as usual proved the most popular of the sports events, although the field sports drew a good number of participants and a big crowd of spectators. Sections stunts were numerous and entertaining this year, the Cleveland, Detroit, Indianapolis, Chicago and Dayton groups all staging one or more stunts. The results of the sports events follow:

Golf: Championship flight, won by Sanford Brown. Flight A, won by Joseph Bijur. Flight B, won by C. E. Salisbury. Flight C, won by F. S. Slocum. Flight D, won by R. S. Drummond. J. H. McDuffee had the low qualifying score.

Tennis: Singles handicap tournament won by Norman G. Shidle (scratch). Doubles won by C. E. Johnson and T. N. Burke.



Col. W. G. Wall, who was chosen to succeed Mr. Hunt as first vice-president of the S.A.E. next year. He was for many years chief engineer of the National Motor Car Co., served the Government with distinction during the war, and is now a consulting engineer with headquarters at Indianapolis. He is the inventor of the Wall oil rectifier

President T. J. Litle, Jr., addressed the semi-annual business meeting and in the course of his talk predicted the coming of smaller cars equipped with superchargers

Trap shooting: Championship won by R. S. Ellis; second, Louis Chevrolet.

Horse Shoe Pitching: Won by C. H. Jorgenson.

Field Events: Fat Men's Race—Won by R. E. Wilson. Three-legged race—Won by Neil McMillan, Jr., and L. C. Porter. Shot-Put—Won by Charles Marcus. 50 yd. dash for men over 30—Won by Neil McMillan, Jr. 50 yd. dash for men over forty—Won by J. C. Talcott. Intersectional Relay—Won by Detroit Section. Inter-sectional baseball—Won by Detroit (by default).

The reports of the various divisions of the Standard Committee, which were reviewed in *Automotive Industries* of May 20, were adopted practically without change. The recommendations regarding the adoption of the Army and Navy aeronautical standards as S.A.E. standards, revised lighting plant ratings and an extension of the lamp specifications were withdrawn by the respective divisions before being presented. All of the other recommendations previously printed were accepted.

In addition to the reports previously printed, a report was made by the tire and rim division, H. M. Crane, chairman, which had held a meeting at Detroit on May 8, at which it was decided to recommend a revision of the present general information on balloon tires. Past standardization in this field covered tires as well as rims, but the proposed recommendation was limited to rim dimensions, the object aimed at being dimensional interchangeability of tires. It was stated that standardization of tire dimensions for low pressure tires would be impossible under present conditions.

Report is Tabled

The recommendation was in the form of a table of pneumatic tire rims for low pressure tires which specified nominal rim diameters of 21 inches together with nominal rim width of 3½, 4, 4½ and 5 in., and a rim diameter of 20 in. together with a rim width of 4½ in. C. C. Carlton, of the Motor Wheel Co., opposed the adoption of this recommendation on the ground that several important changes in practice were impending, and upon motion it was decided to lay the report of the division on the table.

The tire and rim division held a meeting the same day and decided to include a 20 by 4 in. rim in the list. This proposal was presented to the Society at a later session and the recommended list was then adopted.

Racing—

Supercharged, Two-Cycle Engine Opens Up New Field in Passenger Car Design

By Walter L. Carver

NEW possibilities in the passenger and commercial car fields are opened up by the application of the supercharger to engines of two-cycle design.

If the valve troubles of the conventional four-cycle engine can be eliminated there is no immediate limitation upon the speed of crankshaft rotation.

Racing engines of today deliver practically 1.5 hp. per cubic in. displacement.

Development of knowledge of alloy steels and their heat treatment is largely responsible for the outstanding successes in both racing and passenger cars.

These were some of the high spots of the race session which was enlivened by the presence of America's two foremost designers of racing engines and cars, Fred Duesenberg and Harry Miller. In addition to these men, participants in a more or less informal meeting included Ralph Teetor, chief engineer of the Indiana Piston Ring Co., Thomas Midgley, Jr., of the Ethyl Gasoline Corp., William S. Harley, of the Harley-Davidson Motorcycle Co., and a number of others who have been connected with the development of high speed engines and racing practice.

Following closely upon the Indianapolis race as the meeting did, the air was full of talk of superchargers, two-cycle engines, front drive engine speeds, valve trouble and almost the whole gamut of racing terminology. At every point, the discussion hinged upon the effect of the lessons of the race upon passenger and commercial car development. Fred E. Moskovics, president, Stutz Motor Car Co., who has been in touch with the racing game for many years, presided. Men engaged in every phase of automotive activity took a lively interest and the meeting consisted chiefly of questions directed at Mr. Duesenberg and Mr. Miller.

Harry Miller (left), famous designer of racing cars, was one of the prominent speakers at the race session. Right: Fred S. Duesenberg, also a leading designer of racing cars, whose talk at this session met with wide-spread interest

Describes New Engine

At the outset, Mr. Duesenberg described his new two-cycle racing engine in considerable detail, stating that the engine on the track in the recent race fell short of the full possibilities of this type due to lack of time for sufficient experimental and development work. A general discussion of this engine appeared in *Automotive Industries* of June 3. As entered in the race, the porting on both exhaust and intake sides was not sufficiently large, causing a reduction in speed which reduced the supercharger pressure to 9 lb. As against a desired speed of around 6000 r.p.m., only 4800 r.p.m. was possible. The desirable supercharger pressure is 16 lb. which can be obtained when the engine runs at the desirable speed.

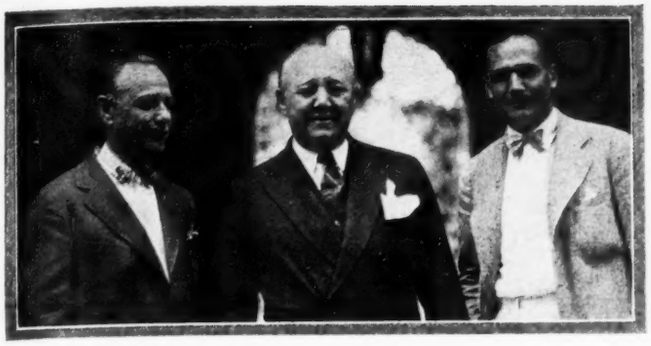
At present the ports in the engine are ½ in. high. It is expected that the increase of the height of the intake ports to 1⅛ in. and the exhaust ports to ⅞ in. will bring the crankshaft speed up to at least four-cycle practice. In the discussion it was stated that Fiat has built an eight-cylinder opposed two-cycle engine equipped with a supercharger which is capable of 11,000 r.p.m. of the crankshaft.

Due to a slight difficulty arising in the pressure lubrication of the rotary valve in the Duesenberg engine, oil consumption at present is somewhat high and causes smoke at the exhaust. However, this trouble is being corrected. In spite of relatively poor pressure conditions within the engine during the race, the fuel consumption compared very favorably with the four-cycle engines. No oil scraper rings were used and no difficulty due to crankcase explosions was experienced.

Superchargers, speed and power characteristics formed a large part of Mr. Miller's discussion in reply to questions. The eight-cylinder engines of 2 3/16 in. bore and 3 in. stroke which were in the race have the following horsepower characteristics:

r.p.m.	hp.
7000	154
6500	150
6000	136
5500	118

At 7000 r.p.m. of the crankshaft, the peripheral velocity



F. E. Moskovics (center), president of the Stutz Motor Car Co. of America, presided at the race session, which was one of the liveliest of the meeting. Among the interested attendants at the session were C. A. Musselman, president, Chilton Class Journal Co. (left), and George A. Hoeveler, president, Hoeveler-Stutz Co., Pittsburgh, (right)

of the supercharger rotor is 68,500 f.p.m., the diameter being 7 in. Valves in these engines are 1 3/32 in. diameter in the clear and the lift is 5/16 in. The range of valve spring pressures from the open to the closed position of the valve is 110 to 87 lb. Mr. Miller states that the valves and their operating mechanism are practically the only limitation of engine speed.

On the supercharger subject, both men agreed that the power requirement estimate of some time ago is erroneous and that from 15 to 20 hp. are required to drive this unit at top engine speeds. Both showed a preference for the centrifugal type supercharger, although Mr. Duesenberg believes the Roots blower or some similar type has pronounced advantages when two-cycle operation is desired. This type produces more positive pressure at low speeds and opens the way for more flexible engine performance.

Front Drive Advantages

In the opinion of Mr. Miller, the front drive offers substantial advantages to passenger car application. Unsprung weight of the front drive axle is only 20 lb. greater than that of the conventional front axle while the rear axle is much lighter than either present axle. Wheelbase need not be increased as in the Miller scheme; the transverse change gear assembly tends to keep the projection in front of the engine very compact. Universal joints and short propellor shafts between the differential and wheels run at wheel speeds so that slight variations in angular velocity are of no particular importance. The amplitude of the front springs is small compared to that of the rear and the mounting of the brakes as sprung weight with the frame and power components is considered a pronounced improvement over present conventional designs.

According to both Mr. Miller and Mr. Duesenberg, there will be no sudden trend toward the light, high-speed engine, although the long swing tendency is in this direction. There are many uncontrolled variables in the owner-driver field which are not directly controlled as they are in present racing practice. Mr. Duesenberg told of one foreign engineer previously an adherent of the light, high-speed school, whose next passenger car model will have an engine displacement of about 800 cu. in. and gear ratios of around 3 to 1. Both seem to feel, however, that there is an immediate future for broader application of both supercharger and two-cycle principles.

Design of Racing Rings

The technique of design of racing rings was discussed briefly by Mr. Teetor who explained that this ring is made relatively narrow to cut down friction, that the wall pressure is relatively light but uniform all around.

An interesting sidelight was the spark plug discussion of O. C. Rohde, of the Champion Spark Plug Co. New plugs for use in the Duesenberg two-cycle engine have been developed. These tend to run much colder than the ordinary or previous types of racing plugs. Experienced racing drivers and abservers check the nature of mixture by observation of the spark plug porcelains. A lean mixture leaves the porcelain very white. A rich mixture causes a dark discoloration while the best mixture ratio produces a light tan or buff color.

Motorcycle Race Engines

Mr. Harley told of his disappointing experience with a supercharger for a motor cycle racing engine. The supercharger required more power than the engine was capable of producing. His company at this time builds a 21 cu. in. air cooled engine which delivers 21 hp. at around 7000 r.p.m. Mr. Midgley reviewed the experience of racing drivers with tetra ethyl lead used with a coast gasoline of high detonation point. The proportion of dope is varied in accordance with the compression pressure of the engine. Operation is hampered by the use of an excess amount of the ethyl fluid and by the use of too little.

Tribute to the advance of American knowledge of alloy steels and their heat treatment was paid by W. G. Wall who designed the second winner and has been connected with the technical control of the race for many years. R. W. Brown of Firestone stated that the present racing tires are four-ply with light casing sections. The field of racing tire development will follow two lines according to his predictions; first, compounding of treads to obtain greater resistance to abrasion, and second, the production of a more intimate bond between the tread and the fabric carcass.

During the free-for-all discussion, Mr. Duesenberg stated that he intended to develop a two-cycle engine with the rotary distributor valve in the head instead of alongside as at present. He believes that better operation will be secured and the piston design will be simplified. No trouble due to the action of the heat of explosion upon the sleeve is anticipated.

Left: R. R. Teetor, chief engineer, Perfect Circle Piston Ring Co., who took part in the discussion at the race session. Right: W. S. Harley, chief engineer, Harley-Davidson Motor Co., brought to the race session information gained from his experience in designing race motorcycles

Gears—

Interest Displayed in Descriptions of New Hypoid and Worm Gear Developments

By P. M. Heldt

THE two papers presented at the Gear Session covered new developments in the line of passenger car rear axle drives, and as the systems dealt with are, or are likely to be, in direct rivalry with each other, a spirited discussion might have been anticipated; however, while there was an evident eagerness to learn something about features that had been incompletely explained in the papers, no real debate developed. The worm gear already is well known, while the hypoid gear is a newcomer in the automotive field, and this is said to be the fist time that means for its practical production have been developed.

A. L. Stewart and Ernest Wildhaber in their paper on Hypoid Gears discussed a new type of gearing developed by the Gleason Works and specially adapted for rear axle drives. Hypoid gears resemble bevel gears, but the axis of the pinion is offset from the axis of the gear and does not intersect it. Previous work on hypoid gears has been done by H. D. Williams, G. M. Bartlett, R. M. Uhlmann, Reginald Trautchold and Nicola Trbojevich. The Gleason Works has developed new tooth shapes and new production methods, and these developments are covered by a series of pending patents.

Two general types of tooth shape were developed in succession, the earlier one being formed without generation and the later, which supersedes the earlier one, with generation. The generated gears can be cut on standard spiral bevel gear generators, while the pinions are cut in a special machine differing from the standard generator in having extra adjustments.

Offsets of hypoid gears have been standardized at 1½, 2, 2½, 3, and 3½ in. The direction of offset determines the hand of the spiral; in a rear axle drive a pinion set below the gear center will have a left hand spiral, and vice versa. For such drives it is preferable to set the pinion below the axis, because in that case the end thrust tends to move the pinion away from the gear rather than to wedge it into it, and because the contact between tooth surfaces on the driving side is more intimate. This latter advantage is most pronounced in pairs of moderate ratio, such as would be used for passenger car drives. As far as thrust loads are concerned, while they are greater with a pinion above the axis, they can be readily taken care of in that case, too, by providing bearings and housings of sufficient rigidity.

The offset of the axis is the cause of a sliding motion at the tooth surfaces, and the teeth of hypoid gears must be of such form lengthwise as to permit of this sliding motion. This imposes restrictions on the pitch lines which do not exist in the case of bevel gears. Fig. 1 shows on an enlarged scale the area around a point *1* on the line of action, *2* and *3* being the contacting tangents at point *1*. During a small motion of the gear and pinion the pitch line of the gear, shown here as a straight line, moves in the direction *4* of the gear periphery perpendicular to the projected axis of the gear. Point *1* of the gear then moves to *5*. The pitch line of the pinion moves in the direction of the periphery of the pinion, that is, in the direction *6*, perpendicular to the line of action. Point *1* of the pinion then has arrived at a new position *7*. Evidently the distance *5-7* represents the lengthwise sliding motion.

The rate of sliding could be expressed as a percentage of the peripheral motion, but since the peripheral motions, *4* and *6*, are different, this would result in different values for the gear and the pinion, and it is therefore considered preferable to express it as a percentage of the normal motion *8*, that is, of the displacement of tangents *2* and *3* in a normal direction.

Let h_p and h_g represent the spiral angles of the pinion and gear, respectively, as indicated in Fig. 1, and let S (= 5-7) be the sliding motion and M (= 1-9) the normal motion, then

$$S = (7\text{-}9) - (5\text{-}9) = M \tan h_p - M \tan h_g$$
$$S/M = \tan h_p - \tan h_g$$

It may be pointed out that in hypoid gears the peripheral motions *4* and *6* are at an acute angle to each other. For this reason the sliding motion is comparatively small and the efficiency of the gears is high. This sliding motion, however, in conjunction with the overlapping feature already existing in spiral bevel gears, makes hypoid gears unusually quiet and makes it possible to lap them rapidly and effectively.

It is a well-known fact that with a pair of helical gears

C. D. McCall (left), who presided at the Gear Session. Right: A. L. Stewart, chief engineer, Gleason Works, who read the paper describing the new hypoid gear rear axle drive

with axis at right angles, and with a worm and wheel, the ratio of pitch diameters usually is not that of the respective numbers of teeth. In consequence, the circular pitches are different for the two members of a pair and only the normal pitches are equal. The same holds true for hypoid gears. The circular pitch of the pinion and that of the gear are proportional to their respective peripheral motions, represented by *6* and *4* in Fig. 1. Their ratio evidently is equal to cos. h_g 1 cos. h_p, the inverse ratio of the cosines of the spiral angles. The hypoid member with the larger spiral angle has the larger circular pitch. The pitch diameters themselves are proportional to the circular pitch and to the number of teeth. The spiral angle h_p of the hypoid pinions is being made 35 or 40 deg., depending on the number of teeth in the pinion, while the spiral angle of the gear is kept between 5 and 15 deg.

Pitch Diameter Increased

As a consequence, the average pitch diameter of a hypoid pinion is increased by from 20 to 30 per cent over the average pitch diameter of the corresponding bevel pinion, the gear diameter remaining the same. This naturally increases the strength of the hypoid pinion. Moreover, large ratios, which are difficult to obtain with spiral bevel gears on account of the resulting prohibitively small pinion diameters, are quite practical with hypoid gears. A still larger proportional diameter increase may be obtained by increasing the spiral angle of the pinion. For instance, with a 10 to 1 ratio and a gear diameter of 10 in., the hypoid pinion may be made with a pitch diameter of 1.75 in., while that of a corresponding bevel pinion would be 1 in.

In computing the torque loads of a pair of hypoid gears the tangential force P at the center of the gear face may be determined from the known torque, and the pressure or load normal to the tooth surface is then determined by dividing by the cosine of the normal pressure angle a and by the cosine of the spiral angle h_g of the gear.

$$P_n = P/\cos\, a \cos\, h_g$$

P_n is the total tooth load and is only slightly larger than the effective tangential force P on the gear, for if we give a an average value of 17½ deg. and h_g, of 8 deg., we obtain $P_n = 1.06\ P$, that is, an increase of 6 per cent.

In spiral bevel gears the total tooth load P_n is considerably greater than the tangential force. With a pressure angle of 17½ deg. and a spiral angle of 35 deg. the total tooth load is 1.28 P, that is, the increase is 28 per cent as compared with 6 per cent.

Left: Ernest Wildhaber, experimental consulting engineer, Gleason Works, who collaborated with Mr. Stewart in the preparation of the paper on hypoid gears. Right: L. R. Buckendale, sales and development engineer, Timken-Detroit Axle Co., pointed out advantages of the automotive worm gear

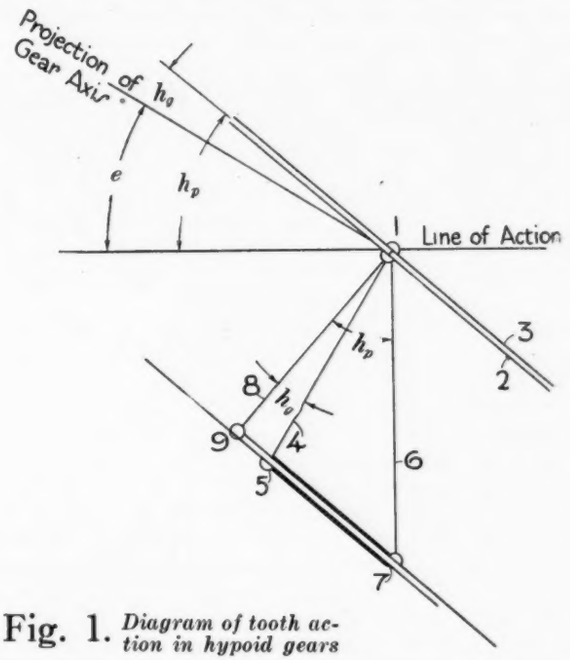


Fig. 1. *Diagram of tooth action in hypoid gears*

In a general way the tooth profiles are similar to those of spiral bevel gears. However, opposite tooth profiles of hypoid pinions are differently curved. The concave tooth side of a hypoid pinion contains a flatter profile than the convex tooth side. Partly on account of this difference in curvature, the different pressure angles of 17½ and 20 deg. are being used on the drive side and the reverse side, respectively, that is, on the concave and the convex sides of the pinion teeth.

The surface stresses in contacting teeth, which are very important factors on account of their decisive influences on the wear and life of the gears, may be computed with the aid of the Hertz formulas which are largely used for computing the stresses in ball and roller bearings. By applying these formulas it is found that helical and straight teeth have practically the same surface capacities, for although tooth contact is more intimate in helical gearing, the depth of the helical teeth is smaller, being proportional to the normal pitch, and the normal or total load is larger on helical teeth for the same torque transmitted.

Total Tooth Load Smaller

Considering hypoid gears, the intimacy of tooth contact is substantially the same as in spiral bevel gears, when taking the average of opposite tooth sides of hypoid gears. But, comparing pairs with gears of equal diameter, the total tooth load on hypoid gears is considerably smaller. Moreover the tooth height, being proportional to the normal pitch, is larger. This explains why hypoid gears are able to carry a considerably larger load under equal surface stresses than either spiral bevel or straight bevel gears. The ratio of load capacities is $(\cos\, h_g/\cos\, h_p)^2$. For spiral angles h_p of the pinion of 35, 40 and 45 deg. and corresponding spiral angles of the gear of 5, 10 and 15 deg. the proportion figures out to 1.48, 1.66 and 1.87. Therefore, theoretically these hypoid gears have 48, 66 and 87 per cent more surface capacity than spiral bevel and straight bevel gears with gears of equal diameter.

In production the gear is cut exactly like a spiral bevel gear of the same pitch angle, while for the pinion a Gleason generator with additional adjustments is used and the pinion axis is offset from the axis of the cradle. The cutter is of the usual Gleason type, having straight cutting

edges at an angle to its axis. Preferably, different cutters are used for the gear and pinion. One gear cutter and one pinion cutter cover all cases. It is not necessary to use cutters with varying amounts of pressure angle correction, as has been the practice with spiral bevel gears.

Conditions of mesh between a hypoid gear and pinion are illustrated in Fig. 2. *A* and *B* are the pitch surfaces of the gear and pinion, these being tangent to a common plane, that of the paper. The projected axis of the gear and pinion are indicated by *C, D*, which also represent the lines of contact between the pitch plane and the pitch surfaces.

The line of action extends along a line *E* which is inclined by a small angle *u* to line *D*. Point *F* may be determined by drawing line *G* connecting points *H* and *I* and by drawing line *J* through point *K* at right angles to line of action *E*. The intersection point *L* is then projected to normal *M*, thus locating point *F*.

The normal pressure angles along points *N* of line of action *E*, that is, the inclinations of the tooth normals at points *N*, can be determined from the known structure of the gear and the cutter which produces it. They change slightly along the tooth and are also different from the pressure angle of the pinion cutter, except at point *K*. To cut a pinion with pressure angles exactly matching those of the gear along the whole length of the teeth, another line of action *O* is determined for the mesh between the pinion and its crown gear, which is represented by the cutter. The mesh during generation therefore extends along a line *O* different from the line of action *E* of the pair of hypoid gears. Line *O* is inclined by a small angle *w* to line *D*. The final step is to determine a crown gear which can mesh along line *O* with the pinions and which contains tooth sides that are conical surfaces of suitable diameter. A tooth side of this crown gear is then represented by a cutter, and the pinion is generated while meshing with the imaginary gear.

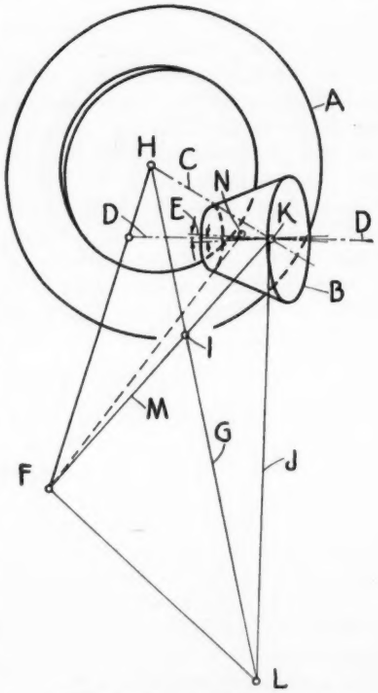


Fig. 2. *Conditions of mesh between hypoid gear and pinion*

By the method developed, tooth surfaces may be produced which match those of the mate gear along the whole length and along the whole depth of the profiles. However, for reasons of adjustability, that is, to provide a certain range of running positions and to allow for non-rigid mounting, the bearing area of the teeth is preferably somewhat restricted, especially in the case of rear-axle drives. Any desired deviation from full bearing may be obtained lengthwise of the teeth and on the depth, that is on the profile. More curved or less curved tooth profiles on the pinion are obtained by changing the offset between the pinion and the crown gear, after refiguring the settings.

If the offset on the drive side is increased during generation, the resulting profile will be flatter and vice versa. The opposite holds true for the tooth side of the pinion, which is on the side of the axis of the gear or crown gear.

Bias bearing, that is a tooth bearing which extends obliquely across the tooth surface, has occurred and has been found objectionable in certain cases of curved tooth gearing. The pressure angles of hypoid gears are made to match along the whole length of the teeth and bias bearing is thus eliminated.

The tooth bearing of a pair of hypoid gears under production can be raised or lowered, or moved endwise, by simply changing the cutter setting. If it is desirable, for instance, in order to counteract hardening changes, to shift the tooth bearing, it can be moved readily, as desired.

Same Production Operations

Production operations are the same in general, for hypoid as for spiral bevel gears. This includes preparing the blanks, cutting the teeth, hardening, grinding bores, shanks, etc., and lapping the teeth. The blanks are turned to different dimensions, being larger in the case of the pinion, and of slightly different face and back angle and outside diameter in the case of the gear. Machines for rough-cutting the gear and pinions and for finish-cutting the gear are the same as used for spiral bevels. The pinion finish cutting machine has additional adjustments for setting purposes, after which the cutting operation itself is the same. The added settings are a vertical adjustment of the work spindle in accordance with the offset of the pinion with respect to the gear, and two angular adjustments of the cutter spindle, about horizontal and vertical axes respectively.

In cutting pinions, it is found advantageous to cut the top side on one machine and the bottom side on another machine, This saves time as nearly all the machine settings have to be changed in going from the top to the bottom side. Pinion cutters are then made with all outside or all inside cutting edges, which permits a speeding up of the machine. Cutters of 9 in. mean diameter are used in all cases of hypoid gears up to date.

It will be seen from the above that production costs of hypoid gears will vary but little from costs of spiral bevels. Cost of blanks will be changed only in the case of the pinion, where the increase in diameter will call for a slightly larger forging. If advantage is taken of the additional load-carrying capacity and gear diameters are reduced, this will more than offset additional cost of pinion blanks. Cutting cost for the pinions will be slightly less on account of using all inside and all outside blades on the cutters.

Influence on Chassis Design

The use of a ring gear of smaller diameter and the location of the rear end of the propeller shaft will have important influence on chassis design. On account of the capacity of hypoid gears for greater transmitted load, a decrease of about 10 per cent in ring gear diameter can be made without increasing unit stresses. This makes possible a greater road clearance. The lowered position of the driving pinion removes the chief obstacle to lowered floor boards at the rear ends. Advantage can be taken of this to lower the whole body.

In the discussion it developed that the straddle mounting involves somewhat greater difficulties in the case of hypoid gears than in that of spiral bevel gears, as the space for the inside bearing is somewhat more cramped. Mr. Stewart pointed out in this connection that the majority of passenger car rear axles in which hypoid gears have been installed so far had overhanging bearings; that is to say, the spiral bevel gear rear axles which

(Continued on page 1004)

With the Sports Photographer at French Lick

Above: Joseph Bijur, president, Bijur Lubricating Corp., came out ahead in flight A of the golf tournament. Center, top: Waiting to drive off from the first tee of the lower golf course. Extreme right: Sanford Brown, director of engineering, Bakelite Corp., won championship flight in golf tournament. Bottom: A golf foursome, left to right: William Fairhurst, sales manager, Spicer Mfg. Corp.; Henry M. Crane, technical assistant to the president, General Motors Corp.; C. W. Keegin, Spicer Mfg. Corp.; W. P. Davidson, General Motors Corp.

Photos
by
Lazarnick

Would you believe it—they're not really bakers! The white hats are to make them visible to the camera's eye when the aerial photographer comes for the human letter picture

Left: Crowd gathered to witness the field sports. Right: The town band of 15 pieces entertained the engineers on the lawn in front of the hotel

(Continued from page 1002)

were to be replaced by the hypoid gear axles had overhanging pinions.

In reply to a question Mr. Stewart said the experience of this company so far had shown them that the same bearings would do for the hypoid gear as would be needed on a spiral bevel gear to transmit the same power.

Chas. Crawford asked how much offset could be used with the hypoid gear, to which Ernest Wildhaber replied that the offset is limited to a little more than one-half the diameter. Mr. Crawford thought this was not enough. The offset is wanted for the purpose of lowering the propeller shaft and thus permit of lowering the floor of the body, and 5 in. of drop was not too much.

The question was asked by M. C. Horine whether the hypoid gear was equally applicable to motor trucks as to passenger cars. Mr. Wildhaber stated that with the hypoid gear it was possible to get a greater reduction ratio than with the spiral bevel gear, hence it could be used for larger vehicles of lower speed than that type.

E. S. Marks asked whether the teeth of the hypoid gear would be more susceptible to distortion during heat treatment than the teeth of spiral bevel gears, but was told experience so far had not indicated that there would be any more trouble from this source.

Mr. Horine thought that if the over-mounted pinion was objectionable on account of lower efficiency, the hypoid would be less suited to truck drives, as in trucks the pinion shaft has to be mounted high on account of the high frame. It was brought out that hypoid gear drives with ratios as high as 8:1 had been built and ratios as high as 11:1 were under consideration while even 15:1 seemed practically possible.

With the pinion mounted above the gear axis there is a tendency to draw the pinion into the gear, and if the mountings were not correct this might break the teeth. It was admitted that this condition obtained, but it was stated that it could be readily guarded against by providing proper mountings which are required also with all other types of gear.

Automotive Worm Gears
—*L. R. Buckendale*

L. R. BUCKENDALE of the Timken-Detroit Axle Co., in his paper on "The Automotive Worm Gear," said the advent of the high speed motor bus made new demands upon the manufacturers of worm-gear-drive axles. Engine speeds and outputs were higher than in truck work, and the reduction ratios needed were lower. Today the reduction ratio in some cases is as low as 4:1. This change in ratios necessitated the development of new methods in production and an intensive study of the principles involved in worm gearing. In bus work the worm is generally undermounted, as it is not necessary to have so great a clearance, and this permits of lowering the center of gravity.

It was found in this development work that the sliding speed which formerly was regarded as the practical limit could be greatly exceeded with safety.

Mr. Buckendale dwelt at some length on the F. J. type of worm gear which was invented by F. J. Bostock. This type of gear came into use about 1918. Previous to that time the involute type of tooth was used exclusively, under which system the sides of a worm tooth section in an axial plane are straight lines, while the corresponding sides of the gear tooth are involute curves. In the F. J. type of worm a straight line generatrix is also used, the same as in the involute type of worm gear, but instead of passing through the axis of the worm, it is tangent to a cylinder of approximately the root diameter of the worm. With the F. J. system, the sides of the worm tooth in an axial plane are convex, while the sides of the gear tooth in the same plane are concave. With this system the conditions of contact are such as to greatly facilitate lubrication. The direction of sliding motion is nearly at right angles to the line of contact, so that a wedge of oil is fed to the contact surface.

Silence Due to Sliding Contact

The silence of a worm gear drive is due to the great obliquity of motion of the surfaces as they come into contact. In other words, they slide into contact, instead of rolling. Experience with motor bus drives has shown that worm drives retain their silence throughout their life. It has been found practical to mount a propeller shaft brake ahead of a worm gear set, which proves the ability of the worm gear to stand shock loads.

One valuable feature of worm gearing is that, by reason of the shape of the tooth surfaces, they can be readily finished by grinding, which permits of obtaining accurate working surfaces after the worms have been hardened. The worm tooth surfaces constitute the surfaces traced by the generatrix for both the worm thread surfaces and the gear tooth surfaces, hence, if the worm teeth are very close to the theoretical form, any wear on the teeth of the wheel will tend to make these teeth approach more closely to their theoretical form. Any imperfections of the worm wheel teeth are therefore automatically corrected by wear.

With all forms of gearing it is necessary to allow a certain amount of backlash, to allow for inaccuracies of manufacture and to provide space for an oil film. This backlash is objectionable because it results in a knock due to torque reversal, particularly at low engine speeds. With worm gearing the tendency to knocking of the teeth under these conditions is practically eliminated, which is believed to be due to the obliquity of the tooth surfaces, the large areas in contact and the low resonant qualities of the materials.

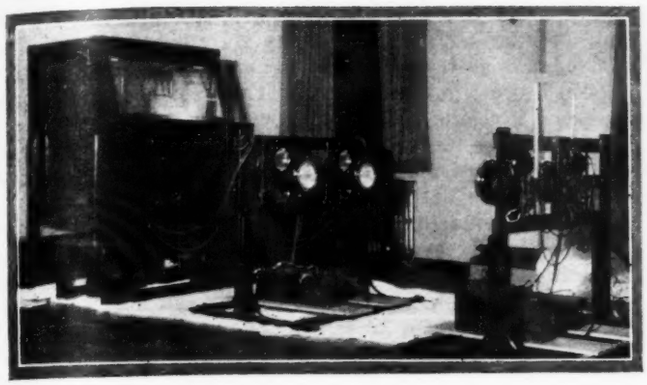
For passenger car rear axles the worm gear drive has the advantage that it permits of lowering the floor of the car and hence the center of gravity, which is an important desideratum in these days when speeds of 60, 70 and 80 m.p.h. are expected.

Discussion of Paper

Discussing the paper on the automotive worm gear, John Younger said he was very enthusiastic regarding the future of this drive for passenger cars. Passenger cars with this type of drive had been built as far back as 1900. At that time the grinding of worms was an unknown art and the steel worms after hardening were merely sandblasted to remove the scale. In spite of this rather crude finishing method, they had a very satisfactory life. Today about 12 per cent of all passenger car models in production in England have worm gear drive.

Otto M. Burkhardt suggested the use of hardened steel for the worm wheel to reduce the danger of abrasion of the tooth surface.

A manufacturer of worm gear bronze gave some data on the properties of this material. A tin bronze is most desirable on account of its good bearing qualities. Some of the high physical qualities of other bronzes are thus sacrificed, but it is sought to compensate for this as far as possible by chilling the bronze. A method of casting the gear rings has been developed whereby chills are placed on three sides of the mold. Bronze gear rings so cast show from 48,000 to 50,000 lb. per sq. in. tensile strength.



K. D. Chambers, of the Light and Knowledge Press (right), gave a demonstration of complementary color headlighting, which was the subject of a paper that he presented at the Headlighting Session. The apparatus used in his demonstration is shown at the left

Headlights—

Car Manufacturers Urged to Take More Interest in Problems of Headlight Design

AT the session devoted to headlighting problems, held on Wednesday evening, at which E. C. Crittenden of the Bureau of Standards occupied the chair, a paper on "Complementary Color Headlighting," was presented by K. D. Chambers. A. W. Devine of the Motor Vehicle Department of the State of Massachusetts read a paper on "Common Troubles With Present Headlighting Equipment," and J. H. Hunt and R. N. Falge gave reports of Divisions of the Headlight Test Steering Committee.

In his paper Mr. Chambers described a new system of headlighting and said the idea was designed to solve the blinding glare problem. It is known as the complementary color headlighting system and makes use of colored lenses and colored light filters in the driver's line of vision.

Each headlamp is oval and contains two paraboloid reflectors. The upper one has an orange glass filter in it, in place of the ordinary clear glass lens; the lower one has a blue glass.

Filter Box in Front of Driver

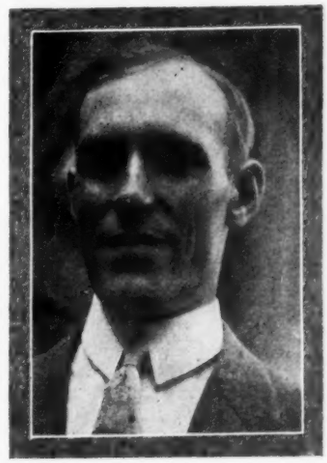
In front of the driver, just inside of the windshield and at the top of it, in the same vertical median plane as the direct line of sight of the driver, is a filter box. This box contains the viewing filters through which the driver looks while driving at night on dark roads. These viewing filters are flat sheets of transparent glass.

The viewing filters are so arranged that when either is drawn down so that the driver is looking at the road through it, the headlights of the same color are automatically turned on. When either filter is pushed up, by means of a knob, the other one drops into the viewing position and the lights are changed automatically. When the bright lights are not in use, both filters are held up in the filter box and are entirely out of sight.

If a car is going north on any main highway, and is conventionally using blue headlights, the driver must be looking through a blue filter. His blue viewing-filter is quite transparent to the light that his headlights are throwing on the path of his car. A car coming south now approaches. On account of the fact that it is southbound, it will be burning its orange headlights and necessarily the driver must be looking through his orange viewing-filter. Consequently, he can see the road as illuminated by his orange headlights.

However, the orange light of the southbound car will not go through the blue viewing-filter of the northbound car and the blue light of the northbound car will not go through the viewing-filter of the southbound car. Consequently neither driver will be bothered by glare from the headlights of the other's car.

Mr. Falge's committee had developed equipment for different light distributions and Mr. Hunt's committee was to make experiments and tests with this equipment. Seven test cars were on hand for the purpose of demonstration. Of these, one was furnished by the Edison Lamp Works and one by the Guide Lamp Co., while several were furnished by the General Motors Co. Most of the cars carried double headlight equipment so as to make it possible to change quickly from one distribution to another. It was explained that this is necessary as the impressions of the quality of road illumination are fleeting and it is necessary to have impression followed



R. N. Falge, engineering department, charge of automotive lighting, National Lamp Works, in collaboration with J. H. Hunt showed headlight test car equipment

quickly by another in order to make a comparison possible.

Mr. Hunt explained that in the past it had been the custom of car maunfacturers to make their choice among the lamp equipments offered on the market without at-

One of the General Motors test cars for demonstrating different light distribution from headlights

tempting to influence the design of such equipments. It was impossible to continue on this basis, owing to the fact that the road lighting problem was becoming more serious all the time and the car manufacturer was held responsible for the equipment which he furnished with his car.

Mr. Devine said he had gathered some information on common troubles with headlighting equipment which was based on a survey they had made in Massachusetts. Troubles generally were due either to the use of inferior equipment or to improper use of the equipment on the part of the operator. Improvement in equipment in the past had been handicapped by a lack of understanding of the requirements.

As an instance of faulty equipment, Mr. Devine mentioned that in one case first a 4-in. disk was used to shut out the light from the center of the reflector and then an annulus which shut out all of the light from the reflector except that from the central 4-in. circle. It was found that there was a material difference in the focal length of the beams with the two arrangements.

Among the chief causes of improper headlighting, Mr. Devine said, were imperfect mounting, wide variation from the designed optical characteristics, inaccurate reflectors and inaccurate positioning of the light source. The question arises. What can the manufacturer do to remedy this state of affairs? The manufacturer, he said, should build his equipment so the operator can comply with the law and, besides, he wished the manufacturer would get the viewpoint of the administrator. All complications in design which make adjustment difficult should be eliminated.

In the discussion H. M. Crane said he was glad the equipment had been brought down to French Lick, as it enabled members to get down to first principles in the headlighting problem. In the past it had not been possible for the operator to make changes in the equipment with a view to eliminating the causes of faulty illumination without at the same time destroying the good points of the equipment. Dr. C. H. Sharp of the Electrical Testing Laboratories said the equipment which had been described in the committee reports seemed to meet the requirements very well, and the next thing was to apply it practically in tests. It was impossible to get too much light on the road for some purposes, and it was a matter of judgment as to what was the best compromise between good road illumination and the permissable glare. The problem was not an easy one but was worth the effort required, for the reason that any solution which might be arrived at would stand until some one came along with something better.

L. C. Porter of the Edison Lamp Works said he was interested in the subject from a somewhat different angle from that discussed at the meeting. The phase that interested him particularly was what happened when the light source got out of focus, as when the filament sagged. At the Edison Lamp Works they had provided an equipment by which it was possible to adjust the light source very accurately.

A question was asked regarding the advantages of spot lights. Mr. Falge said a spotlight would illuminate the right side of the road and was a great improvement from the driver's standpoint over dimmed headlights, but, of course, it did not give as good a driving light as the headlamps when turned on full. A member made the observation that there is much reflection of the light from a spotlight on wet roads, but it was pointed out that since this reflected light does not strike the eyes of other road users it is not important.

Demonstrations after the meeting were given of the complementary color headlighting system in the convention hall where the meeting was held, and of the lighting equipment on the test cars on the roads in the vicinity of the hotel.

Getting Out the Daily SAE!

Getting out the Daily SAE under the direction of the Hyatt Roller Bearing Co. fell to this group of stalwarts and the publication they produced earned the highest praise from everybody at the meeting. Needless to say, very little of their time was spent in the manner indicated by the picture on the right

Fuels—

Industry is Approaching a More Exact Understanding of Fuels and Lubricants

THAT engineers and chemists are approaching a more exact understanding of the nature of both fuels and lubricants and the effects and difficulties which are attendant upon them was the outstanding thought developed by the session devoted to their discussion. Several angles of approach made the scope of the meeting broad with the inclusion of developments from field research, the laboratory of the Bureau of Standards and the refiners of both fuel and oil.

The methods of approach to these factors in motor car operation is indicated by the three papers presented. In the first, O. M. Burkhardt, research manager of the S.A.E., presented a statistical analysis of field observation and analyses made under his direction with the cooperation of more than 100 companies. While this material is not conclusive due to the small number of returns compared with the number of cars in operation, the way to a rather complete understanding of the variables of lubrication and wear or corrosion is indicated.

In the second paper, T. S. Sligh, Jr., of the Bureau of Standards showed a new fuel testing apparatus of the equilibrium type which tends to give refiner and car designer a clearer picture of the nature of the hydrocarbon fuel as related to its operation in the internal combustion engine. In the third paper of the meeting, the carbon producing characteristics of various lubricating oils were discussed by S. P. Marley in a paper prepared by W. A. Gruse, C. J. Livingstone and Mr. Marley, all of the Mellon Institute of Industrial Research. These results are based on engine tests in which the many variables involved were affected singly. The chairman was H. C. Mougey of the research division of the General Motors Corp.

How Samples Were Gathered

Considerable discussion was aroused by Mr. Burkhardt's paper, revolving chiefly around adequacy of the sources of information. Through the cooperation of many companies, oil samples from various parts of the country have been sent in with some attempt, not altogether successful, at a history of the use of the sample. It was hoped to get samples which had run for 500 miles under specific seasonal conditions. These samples originate from various service garages around the country but the possibility of inaccuracy of history and lack of knowledge of the condition and age of the engine from which they are taken complicate the process of drawing conclusions. Further, the number of returns compared to the work required in forwarding the program is discouraging.

An optional method of performing this preliminary work was urged strongly by J. F. Winchester who advocated that these samples could be obtained from owners of fleets who know the life history of the engine, the conditions under which it operates and have records of oil changes. Also he averred that fleet owners, due to their intense interest in engine life and low maintenance,

T. S. Sligh, Jr., Bureau of Standards, and the apparatus by which he demonstrated his talk on "Gasoline Testing" at the Fuels and Lubrication Session

will cooperate to the fullest extent and will supply measurements of component parts. The present method, he insists, is wide open to almost every possible source of error and is bound to introduce misleading factors into any statistical research. While no conclusions as to the relative value of the two methods were drawn in the meeting, general sentiment apparently was in the direction of some combination of the two or even the simultaneous approach to the question through both channels.

Discussing the past tests in some detail, Mr. Burkhardt said in part:

"At present it is frequently recommended that oil should be changed after 500 miles of running. If this ruling were followed we would throw away about 200,000,000 gal. of oil per annum. This figure is arrived at by assuming that 20,000,000 cars when running an average of 5000 miles per annum would change oil ten times."

An investigation of contaminated crankcase oil was directed so as to find answers to the following questions:

1. Is more iron oxide present in the crankcase oil in winter than in summer, i. e., does greater cylinder wear result from winter conditions?
2. Does the kind of oil used make a difference in the amount of cylinder wear?

Left: H. C. Mougey, chief chemist, General Motors Corp. Research Laboratories, who presided at the session on Fuels and Lubrication. O. M. Burkhardt (right), research manager, Society of Automotive Engineers, read a paper on "Causes of Wear and Corrosion in Engines"

3. Does the use of special fuels, such as those containing benzol, result in greater cylinder wear than is encountered when regular gasoline is used?
4. Does the use of rectifying or similar auxiliary equipment decrease cylinder wear when dirt and dust are present?
5. Does the removal of water, by rectifying devices, permit the use of high-sulphur fuel?
6. If water and dirt are eliminated from the crankcase oil, is high dilution objectionable?
7. Does the use of ethyl gas affect cylinder wear?
8. Are some cars worse offenders than others?
9. How frequently should crankcase oil be drained under a given set of conditions?

Motor car engines are subjected not only to normal and abnormal wear but also to corrosion. These phenomena are in turn influenced in various degrees by a large number of conditions that could not be controlled in a survey of this kind. Some of these influential factors are given in the following list:

1. Make of engine—Structural differences, such as bearing clearances, efficiency of the lubricating system, quality of workmanship, and the hardness and other physical characteristics of the bearing surfaces.
2. Age and condition of engine—New engines may show a higher rate of abrasion and more metal particles in the oil than well-worn-in engines.
3. Service that car is rendering—Cars employed in light work will not, if other conditions are alike, show as much wear as others that are used for heavy-duty purposes.
4. Care and attention given to the engine.
5. Quality of lubricant.

Samples collected in the Spring of 1925 were carefully indexed as they arrived at the Bureau of Standards. To reduce the number of analyses and to make samples representative of general rather than individual conditions to which engines may be subjected, oil samples derived from cars of the same make and of the same locality were compounded.

Wear is a function of the weight of the silica, the viscosity, the weight of abraded metal particles, and the number of starts. Starting frequently involves rubbing on dry and partly corroded surfaces.

Corrosion is a function of water, the sulphur content of the fuel, intermittent running, and time.

Temperatures that prevailed during the period when the 1926 samples served as lubricant are from 26 to 49 deg. Fahr. and a higher relative degree of corrosion is shown.

By comparison it appears as if the oil filter and the air cleaner reduce the ash or more particularly the contaminating ingredients of the lubricating oil that are found in the ash to about half the amount that is usually found if this equipment is not used.

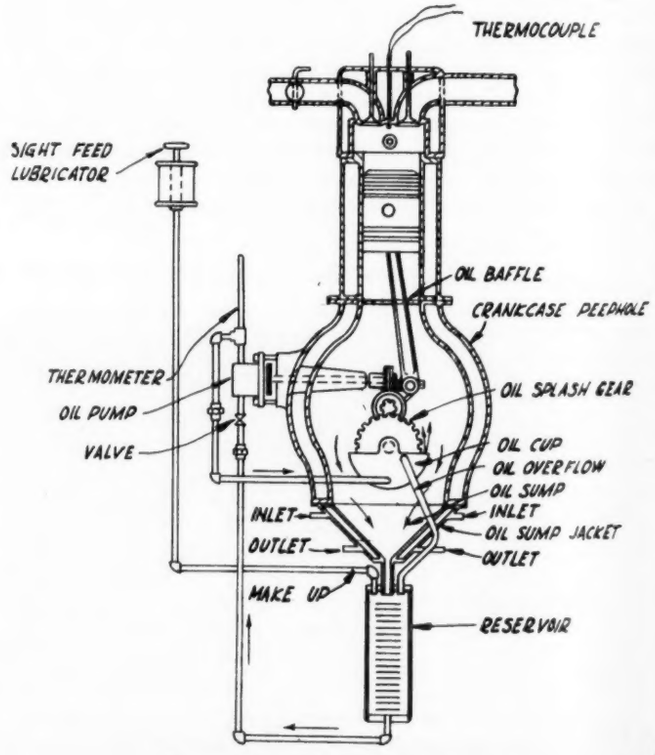
Rectifiers Eliminate Water

Cars equipped with rectifiers showed the presence of very little water in the crankcase. Those equipped without this device or thermostats or shutters indicated more than six times as great water content in the crankcase oil while those having only the two latter devices showed a water content about equal to those having no rectifier.

Talking on "Volatility of Automobile Fuels," Mr. Sligh, Bureau of Standards, said that volatility of hydrocarbon fuels varies with the vapor pressure and temperature. Gasoline of today is a complex rather than simple hydrocarbon and therefore has a changing degree of vaporization for every change in these conditions. The amount which can be evaporated depends upon the adjacent volume and pressure of air or weight of air. In the older or Engler method of checking fuels. no account is taken of fuel to air ratios.

A new type of equilibrium device in which fuel and air ratios can be established was shown by Mr. Sligh. Characteristics of existing fuels show some differences in the two processes. Briefly, Mr. Sligh's instrument consists of a heating element in which is enclosed a compound coil, one part of which introduces a measured flow of air and a section of larger diameter introduces a measured flow of fuel. In equilibrium at any pressure, a portion of the mixture is evolved as vapor, the balance, which is collected as a liquid in a trap, constitutes heavier ends which must be volatilized at higher temperatures. The ratio figures obtained by this process are used to draw the characteristic curve of the fuel.

In the equilibrium apparatus, much of the fuel will



Set-up for carbon deposits tests

Left: S. P. Marley. Right: C. J. Livingstone and W. A. Gruse (in the order named). All are connected with the Mellon Institute of Industrial Research, Pittsburgh, Pa., and they collaborated on the paper which Mr. Marley read at the Fuels and Lubrication Session

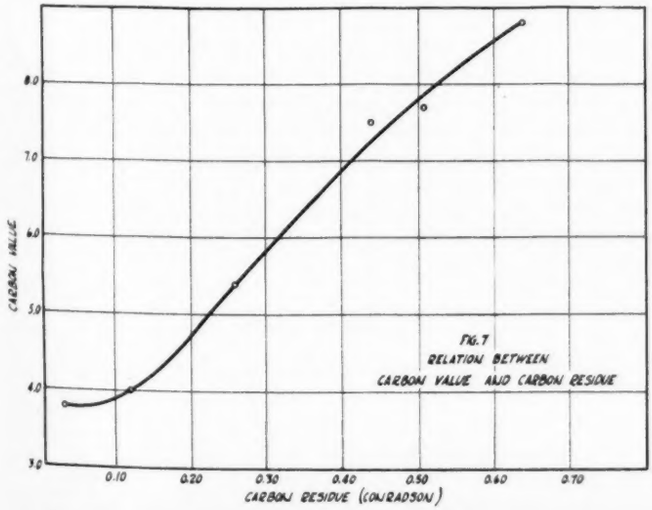
evaporate at 32 deg. Fahr. and the greater portion goes over at 150 deg. Mr. Sligh explained that his instrument is not yet developed to the point where engine conditions can be fully approximated. Pressures in the intake manifold vary tremendously and when the pressure is reduced by half, twice as much fuel evaporates at the same temperature. Under loaded running conditions, fuel evaporization is no problem within the available range of mixture ratios. However, the volatility of the lower ends is highly important for starting, particularly in cold weather. The equilibrium apparatus comes nearer to giving the refiner a conception of this characteristic in terms of the engine performance.

The latter points were confirmed by O. C. Berry of the Wheeler-Schebler Carburetor Co., Inc., who stated that refiners should make a greater distinction in the lower ends of summer and winter fuels.

Paper on Carbon Deposits

The subject of the paper by Messrs. Marley, Livingstone and Gruse was "The Influence of Temperature, Fuel and Lubricant in Forming Engine Carbon Deposits." In studying the effects of these factors, one at a time, all other conditions were held constant, and the authors said they experienced no difficulty in duplicating their results, which previous experimenters in this field had been unable to do.

Contrary to expectations, the carbon deposit with natural gas was only 5 per cent less than with commercial gasolines, and the authors draw the conclusions from this observation that ordinary commercial gasolines when used in fuel mixtures of 12:1 contribute little to the carbon deposit. High values were obtained with naphtha and kerosene and are believed to be due to incomplete oxidation and asphaltization of the sprayed liquid.



Relation between Conradson value and amount of carbon deposited

A series of tests was run with fuels of different degrees of volatility, ranging from natural gas through commercial gasolines to heavy naphtha and kerosene. There was practically no difference in the amount of carbon deposited with the different commercial motor fuels, but the deposits increased when naphtha was reached. A slightly higher carbon deposit was obtained also with a benzol blend, but it was softer and more soot-like than the deposits from the other fuels.

In studying the effect of different grades of lubricating oil, the engine was run for periods of 36 hours at a comparatively low temperature, at which time the dilution equilibrium point had been passed. The mixtures used (of commercial gasoline) were comparatively rich. It was found that Pennsylvania distilled oil and Mid-Continent blend gave greater carbon deposits than a Coastal oil, the viscosities of all three grades at 100 deg. F. being the same. With the same fuel and otherwise under the same conditions the dilution of the different oils was the same.

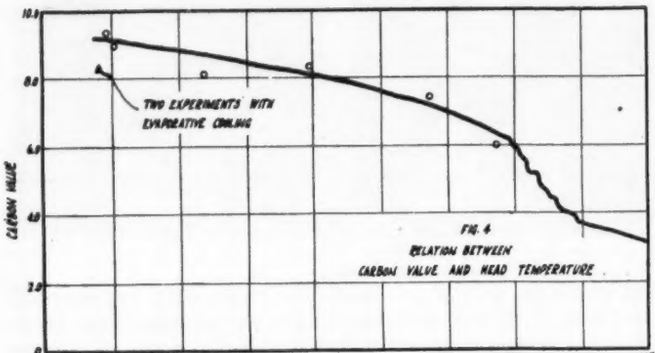
In another series of tests, simulating summer running conditions and using a lean mixture, a Pennsylvania blend oil gave twice as much carbon as a Gulf Coastal oil, and a Mid-Continent distilled oil about 50 per cent more than the Coastal oil. The Gulf Coastal oil was found to be more volatile than the other grades and this is believed to be the reason for the smaller carbon deposit with it. This also is believed to explain the difference in the character of the carbon deposits. The carbon obtained with the Coastal oil was in general dry, powdery and friable over the hot metal areas, and only slightly more coherent over the cooler areas. The carbon from the Pennsylvania oil, on the other hand, was extremely hard and adherent over the hotter areas and sticky and asphaltic in appearance over the cooler areas.

Methods of Avoiding Carbon

It was also found (as shown by the accompanying graph) that the Conradson carbon residue of lubricating oils is an approximate measure of their carbon-forming tendency. The authors make the following suggestions for the avoidance of trouble from carbon: (1) The oil consumption should be kept at the lowest point con-

sistent with a fair margin of safety. (2) Excessive and irregular cooling should be avoided. (3) The fuel mixture should be kept on the lean side of smooth operation, rather than on the rich side. (4) In choosing from a class of oils of the same viscosity, that oil will probably be the more satisfactory which is the more volatile.

The tests were carried out on a modified Delco-Light



Relation between cylinder head temperature and amount of carbon deposited

850-watt generator set engine. The engine was fitted with a water-cooled head having a thermo-couple set into it at the center, and a water-cooled cylinder and the original air-cooled cylinder could be used interchangeably. The crankcase bottom was replaced by a water-cooled sump, as shown in the set-up diagram herewith.

As shown in the drawing, a system was installed whereby a small charge of oil could be circulated, which made possible observations of small variations in oil consumption. In presenting the results, only the carbon deposited on the cylinder head is recorded, one reason for this being that the carbon on the piston was generally mixed with oil.

In the tests in which the lubricating oil was varied, in addition to weighing the carbon on the cylinder head alone, the total amount of carbon on head and piston was also determined and a correction made for oil content of the piston carbon, but the same conclusions as to the effect of the oil may be drawn from both sets of figures. The lower the operating temperature, the softer and less adherent was the carbon deposit.

Two tests were made for carbon deposits when the engine was provided with an evaporative cooling system. It is of some interest to note that (as shown in the graph) the head temperature was no higher than with water cooling and the carbon deposits were the same as with water-cooling giving the same head temperature.

Discussion took some issue with the idea of scraping off carbon deposits for the determination of their value. The idea of photographic and plaster cast records and analytical purposes was advanced. Two speakers stated that more than the characteristics of the oil were involved in carbon deposition, that mechanical fits, conditions of operation, etc., have a profound influence. Also that the refining of the oil rather than its point of origin is of the more value in producing low degrees of carbonization.

Easy Riding-

Better Cushion Springs and New Means of Measuring Riding Comfort Discussed

TWO different channels of approach to the solution of the problems of riding comfort featured the riding qualities session. In the first paper, comfort as it concerns the relationship of the passenger to various road conditions was broken down into its component elements. This paper, "Cushion Springs and Riding Qualities" presented by W. C. Keys, drew the conclusion that of the four elements of primary importance to comfort—tires, springs, shock absorbers and cushion springs—the latter have been neglected relatively. A number of recommendations based upon experience and experiment were given considerable attention.

From an entirely different angle, that of instrumentation which eliminates human guesswork and psychology from the determination of riding comfort values, R. W. Brown described a number of new developments and improvements in this field. His paper, "Instrumentation and Results of Riding Qualities Tests," carries this subject forward from the material which he presented at last year's summer meeting. The first paper was illustrated by a number of samples, while Mr. Brown showed slides of the new instruments and graphs produced and in addition had several instruments at the meeting. About two hundred were in atendance and the session was conducted by J. W. White.

Mr. Keyes' Paper

THE chief function of seat and back cushions is to conform to the contour of the passenger's body and to soften the action of the car resulting from passing over uneven road surface, said Mr. Keyes of the Gabriel Snubber Sales & Service Co. Instead of placing most emphasis on these two features, many manufacturers have given more attention to cost, style and appearance with the result that present day auto-

mobile seats are not as comfortable as they might be.

The seat back cushions are also called upon to cushion the thrusts resulting from a pitching or galloping car. Seat cushions are called upon for a great deal of cushioning of shocks due to car movements. Much of the unpleasant features of usual cushions are caused by rebounds after the initial shock has been received.

A good way to control this rebound is to restrict the flow of air into the interior of the cushion after the springs have been deflected. If the outside material of the cushion is leather or some other substance nearly air tight such a condition is easily obtained merely by providing check valves which permit the air to be expelled from the cushion with little hindrance but permit it to re-enter at a much slower rate.

When porous material is used for upholstery it is possible to line the cushion with oil cloth or other impervious material or even to encase the coil springs in individual envelopes of impervious material and equip each with a small check valve.

With the advance in riding comfort that has followed the adoption of low pressure tires and devices for checking rebound of car springs it is another logical step to develop and adopt better cushioning devices for the seats and backs of all motor vehicles.

Friction Control Suggested

In the discussion of this paper, the possibility of some type of connection between the seat and chassis springs was suggested as a possibility, although the mechanical complexity was fully appreciated. W. G. Wall advanced the benefits of some kind of friction control in the seat spring assembly which will tend to produce easier riding.

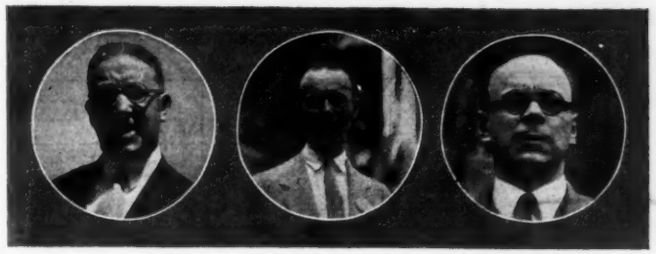
Mr. Brown, in his paper, stated that since the inception of interest in the solution of the problems of riding comfort, one of the great drawbacks has been the lack of instrumentation which interprets values free from errors of judgment and observation. The electric contact type of instrument has been demonstrated as a very satisfactory answer to this need. Since the paper of a year ago, a number of developments which bring this type of instrument with its appurtenances to a high degree of usefulness have been completed. In addition, the filing machine which once was an important factor in the calibration of these instruments has been replaced by a refined reciprocating machine in which speed travel and time factors can be varied without errors due to lost motion, friction, etc.

New types of accelerometers can be applied to practically every point on the vehicle for the purpose of recording the rate of instantaneous acceleration. Apparently this factor to a larger extent than distance or amplitude of travel produces sensations of comfort and discomfort. With the new accelerometers recording chronographs and similar instruments which make records of runs available for analysis and comparison have been developed to a comparable extent. Among the instruments illustrated were the following:

1. Single Element Contact-Type Accelerometers. These have been developed separately by Dr. H. C. Dickinson, Benjamin Liebowitz and the author and are calibrated alike. The same results are produced when the same accelerations are measured.

2. Improved Experimental Contact Accelerometer. Can be used singly or in multiples in which each element is susceptible to a different degree of acceleration. Among the instruments demonstrated was a ten element unit enclosed in an aluminum case permitting application to any point on the car from the axle upwards.

In conjunction with this device a new accelerometer chronograph of the electrical type records the action



Left to right: J. W. White, chief engineer, Wire Wheel Corp. of America, who presided at the Riding Qualities Session. W. C. Keyes, Gabriel Snubber Sales & Service Co., who read a paper on "How Do Cushions and Cushion Springs Affect Riding Qualities?" R. W. Brown, Firestone Tire & Rubber Co., whose paper was on "Instrumentation and Results of Riding Quality Tests"

of each of these units in conjunction with records of time, speed and other functions which may be desirable for individual cases. Another new instrument is a magnetic counter which has been made from a stop watch.

The new calibration machine has a main shaft which carries two heavy flywheels and drives a Scotch cross head mechanism for the purpose of producing various vertical accelerations. The cross head assembly is adjustable to strokes of 4 in. High pressure oiling is applied at all bearings to eliminate lost motion and friction. Speed is changed by a variable speed electric motor.

A number of charts and graphs showing the results produced by these instruments were shown. From these Mr. Brown advanced the folowing conclusions:

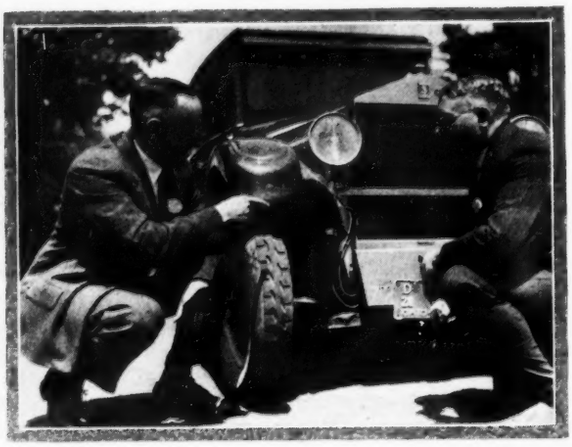
1. At low speeds (0-12 m.p.h.) on average roads, axle accelerations caused by solid tires are generally not in excess of the accelerations caused by pneumatic tires at higher speeds.
2. Axle accelerations are caused by road irregularities and increase rapidly with increase of road roughness. Hence if roads are made and maintained sufficiently smooth, solid and cushion-tired vehicles can be operated economically at higher speeds.
3. Increasing the sectional area of pnuematic tires, with corresponding decrease in inflation pressure, decrease axle displacements and accelerations.
4. In the absence of definite information regarding the physical and mental effect of vibrations, it may be assumed that decreasing displacements and accelerations increase riding-qualities.

J. A. C. Warner, manager, Meetings and Sections Dept., S. A. E., was caught here in a pose which he assumed but rarely throughout the meeting

Men and Events at the Summer Meeting

Vincent Bendix, president, Bendix Corp., had a demonstrating car at the meeting and in this picture the photographer has caught him pointing out to friends what he considers a good front wheel brake

Louis Chevrolet, former noted racing driver, was among those present



E. S. Marks and S. R. Castor of the H. H. Franklin Mfg. Co., are interested in the Sizaire spring axle

Earl Gunn, chief engineer of the Light-Six Division of the Nash Motors Co.

"Here they come!" One of the special trains arriving at French Lick Springs with members of the S.A.E.

Photograph of "Human Letters"

THE above is a reproduction of the aerial photograph taken from an Army Air Service plane at French Lick Springs. The letters "S A E" spelled out on the lawn of the hotel are composed of members of the Society. The performance was given to demonstrate the army's process of quick work photography. Ten minutes after the picture was taken a print of it was dropped to the members on the ground. The photographic apparatus used and the airplane in which it was installed are shown in an accompanying photograph. Major J. F. Curry, commanding officer at McCook Field, who consented to the giving of the demonstration, assigned the piloting of the plane to Lieut. George W. Goddard (right), and the handling of the photographic work to Dr. S. M. Burke (left), respectively chief and physicist of the aerial photographic branch of the engineering division of the Air Service

Left: O. A. Parker with his new hollow-spoke steel wheel

Right: C. H. Foster, chairman of the board, Gabriel Mfg. Co., took a keen interest in the activities

Tires—

Relationship of Tires to Economics of Car Operation Receiving More Attention

TIRES and their relationship to the economics of vehicular operation, road design and maintenance and automotive design are emerging from the category of "just tires" and are becoming engineering entities along with the more mechanical portions of the car. At the same time, new fields of tire design which affect their application and maintenance are growing with a possible chance of complicating further a situation which already is at least rather complex. These were some of the high points of the Tire and Rim Session.

Impact tests made at the experimental station of the Bureau of Public Roads at Arlington, Va., which were presented by J. A. Buchanan and J. W. Reid in a paper, demonstrate that with worn solid tires, the impact shock at the road may reach 9 times the static load of the truck. This tremendous shock is responsible for a serious economic loss arising in worn-out highways and those under repair. Cushion and pneumatic tires afford ready reduction of this loss.

In the latter portion of the program, the association of the balloon tire and the drop center rim was discussed by B. J. Lemon of the Morgan & Wright division of the United States Rubber Co. This type of rim has been introduced to the public by Ford and considerable interest in its future possibilities is evinced in the motor car field, particularly as the drop center or well-base rim has become almost universal in the British market. The session was in charge of C. M. Manly.

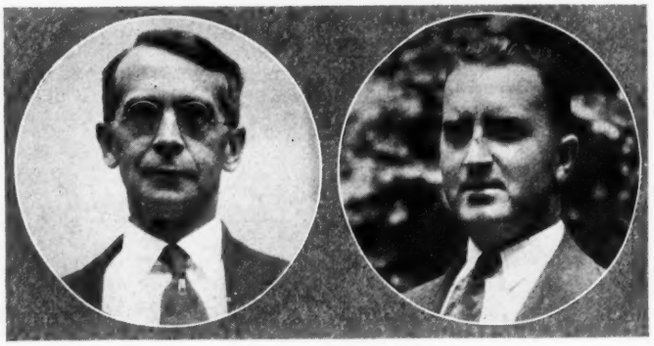
Truck Impact Tests —*Buchanan, Reid*

IN their paper on the general results of the cooperative motor truck impact tests, Mr. Buchanan and Mr. Reid stated that the tests, which are being conducted by a committee representing the U. S. Bureau of Public Roads, the S. A. E., and the Rubber Association of America, have for their object:

1. Determination of cushioning effect as influenced by the various tire truck and road factors.
2. Determination of road stresses as influenced by motor truck wheel impact.

As a result of the tests already conducted the following preliminary deductions were drawn:

1. As static load increases, road impact reaction increases.
2. As static load increases, ratio of road impact reaction divided by static load decreases.
3. Thickness and narrowness of tread rubber are desirable in reducing road impact reaction.
4. Increasing the thickness or profile height of rubber has a very marked effect in reducing road impact reaction in both single and dual mountings.
5. In tire equipments, all of which were standard at the time of the tests, dual mounting causes heavier impact forces than the corresponding single mounting of the same total load-carrying capacity. (This was determined on a pneumatic-tired 2-ton truck and on a solid-tired 5-ton truck.)
6. Appreciable variation of cross-sectional rubber, or breaks in its continuity, causes heavy repeated impacts on the road.



C. M. Manly (left), a past president of the Society, presided at the Tire and Rim Session and also was responsible for outlining to the council a plan for revision of the constitution of the Society. Right: J. W. Reid, Rubber Association of America, co-author of the paper on "Cooperative Motor Truck Impact Tests"

7. Dual-mounted tires should always be mounted with the tread designs staggered.

The tests of tire-cushioning qualities are divided into two general groups, the static tests and measurements, and the impact tests. It is expected that, in most cases, a general relation will be found between these two methods, shortening and standardizing future tests.

In the static tests the tires were placed in a testing machine and vertical deflections were recorded as the load was slowly applied, from which data loads vs. deformation curves were drawn. The areas of tire contact were also taken at three loads on each tire, and reproductions were made of the impressions. From this, the area, width, and length of contact were obtained at the three loads. The hardness of the tread rubber was measured by a Shore "Durometer."

In the impact tests the total vertical road reaction is determined by a specially constructed measuring apparatus known as an accelerometer, which is mounted on the test trucks. This accelerometer was designed to measure the maximum vertical accelerations of the truck wheel and to simultaneously record the proportional deflections of the corresponding truck spring. The forces are computed by formula and may be resolved into

sprung and unsprung components. The sprung component is determined by measuring the change of truck spring deflection from its static position at a given load to its position at the instant of impact. By referring to the proper portion of a previously prepared calibration curve for the truck spring, the change in spring pressure from that at static load is obtained. The unsprung component is determined by measuring the acceleration or deceleration of the truck wheel, knowing the mass of the unsprung parts.

The corresponding total force is computed according to the formula:

$$F = m\ (a + g) + P.$$

wherein F = vertical road reaction in pounds.
m = mass of unsprung truck parts in poundals.
a = acceleration of unsprung truck parts in ft. per sec. per sec.
g = acceleration of gravity, 32.2 ft. per sec. per sec.
P = truck springs pressure in pounds at instant of impact.

The accelerometer is mounted to follow the vertical movement of the right rear wheel of a truck by means of a tight, self-aligning bearing, and the instrument slides on guides on the truck body. The sensitive element is composed of a calibrated coil spring supporting a weight of known mass. A rod is fastened to this weight and acts as a guide and stylus holder. The stylus records the relative movement of the weight on sensitive paper which is rolled on a drum driven by the truck wheel, the paper and truck speeds being therefore proportional. A back-stop is arranged above the accelerometer weight to prevent its upward movement beyond the position at rest and no initial compression is on the spring other than that due to the weight itself.

Characteristics of Accelerometer

The characteristics of this accelerometer are such that the displacement of its weight as recorded on the sensitized paper is proportional to the acceleration of the truck wheel. The calibration factor depends on the characteristics of the accelerometer spring-weight combination and varies with the characteristics of the impressed acceleration from the truck wheel. The accelerometer spring-weight characteristics are accurately determined and that combination is selected which is best suited to the accelerations to be measured.

The characteristics of the impressed acceleration are influenced by the natural characteristics of the tire equipment and the unsprung truck weight in the case of the wheel dropping on a plane surface. In the case where the wheel passes over a relatively narrow obstruction at comparatively high speed, there should be an added correction dependent on the time duration of this condition of shock. Calibration factors for various accelerometer spring-weight combinations have been determined for use with the full range of tire equipments included in the tests.

Left: J. A. Buchanan, Bureau of Public Roads, who collaborated with Mr. Reid on the truck impact tests paper. Right: B. J. Lemon, automotive contact engineer, Morgan & Wright Plant, United States Rubber Co., talked on "Balloon Tires for Drop Center Rims"

Seven Makes of Tires

The tires used in these tests were furnished by seven American manufacturers. The types used were pneumatic, cushion, heavy-duty cushion, and high profile non-skid and regular types of solid tires. Tests have been made with new and worn-out tires, and with tires cut to various heights. The pneumatic tires were tested at standard and three-fourths standard air-pressures. The sizes varied from 32 in. x 3½ in. to 40 in. x 14 in., and the rated carrying capacities varied from 1300 to 11,200 lb. per tire. They were used in single, dual, and tandem mountings, and as overloaded and oversized tire equipment.

The trucks used represent four American makes, the capacities of which are one, two, three and five tons. Loads were applied up to 150 per cent of the tire-carrying capacity, using the types and sizes in standard equipment. The pay loads were 100-lb. weights which were placed within the truck body in a definite manner and according to accepted practice in load distribution. The wheel loads were measured with portable road scales, which were frequently calibrated. In addition, a number of loads were sent to accurate platform scales for checking.

The trucks were driven at speeds varying from 3 m.p.h., at 3 m.p.h. increments, up to the maximum speed obtainable under the load and test conditions. Speeds were determined by timing measured runs with a stop-watch, and were checked for variation by the truck speedometer. Any run which showed a variation of more than ½ m.p.h. from the average speed was discarded, that test being re-run. Before use in the tests the rear truck springs were removed, cleaned, greased, calibrated, provided with jackets, and replaced. For reference, weight distributions of the empty trucks, truck measurements and specifications were recorded.

In order to include actual road constructions, two 500 ft. lengths of nearby highways were selected as test road sections to represent extremes likely to be encountered by through commercial traffic. One of these is an example of smooth concrete construction, and the other is an old stone-block pavement. A 16-wheel profilometer showed an average cumulative vertical variation of 2.5 in. per 100 ft. on the smooth concrete section, and of 18.2 in. per 100 ft. on the rough stone block section. Both sections are straight and substantially level, and have but slight crowns. Tests, in general, on these two sections were confined to tire capacity and 1½ tire capacity loads at a representative speed of 12 m.p.h. The number of impacts (i. e., vertical road reactions) produced and the magnitudes thereof were determined.

Road Specially Built

To provide for duplication of test conditions, a very smooth, straight, and substantially level concrete road was specially built for these tests, and on it various steel obstructions were set in grout and secured by bolts. These obstructions included 30-in. inclined planes with rises of 13/16 in., 1½ in., and 1 15/16 in. respectively; 3-in. rectangular (cross-section) blocks with heights of 9/16 in., ⅞ in., and 1⅛ in., respectively; and rounded (segmental) blocks having 3-in. chords or bases and

heights of ¾ in. and 1½ in., respectively. These obstructions were about 4 ft. wide and placed at about 30 ft. intervals along a painted guide line on the road.

Drop Center Rims

—*B. J. Lemon*

MR. LEMON, of the United States Rubber Company, in his paper on "Balloon Tires for Drop Center Rims," stated that although the balloon tire is now in its fourth year no one as yet seems to have made a serious attempt to define it. About the only feature of balloon tire practice that has been at all permanent is the designation of sectional measurements by decimals. There is much variety in all other respects and the tire industry now apparently faces another departure in the shape of balloon tires for drop center rims.

For automobile use the drop center rim was impractical until the flexible balloon tire came into use. With the advent of this tire in 1923 the Dunlop Company, of England, reintroduced the drop center rim in combination with the detachable wheel for balloon tires.

The slowness of the development of drop center tires and rims appears to be due to three main causes: (1) The flat base rim with the straight side tire has been in use only a comparatively short time, particularly on light cars, and is considered a fairly economical and satisfactory combination. (2) Other problems of vehicle design and improvement, such as changes brought about by balloon tires and four-wheel brakes, appeared more pressing. (3) Only one outstanding American manufacturer has featured the mechanical or sales advantages of the drop center tire and rim combination.

The 4.40 balloon tire for the Ford steel-spoke, drop-center wheel is now made to fit satisfactorily also on the standard flat base 3½ in. straight-side rim, and is therefore called a combination casing. Essential modifications required in the standard 4.40 in. casing to adapt it to the drop-center rim are a bead width not greater than the rim ledge width, a soft feather edge toe to prevent tube pinching, and a somewhat snugger fit to the rim than on flat base rims.

The problem of bead width is not a serious one where cable wire bead construction is used. With the braided wire bead, however, greater construction changes must be embodied to produce a bead width satisfactory for the drop center rim. The narrow bead ledge of the drop center rim presents added bead construction difficulties when, for example, a 4.75 or a 4.95 tire is used as oversize on a 4.40 rim. The beads of such oversize casings must be no wider than those of 4.40 casing.

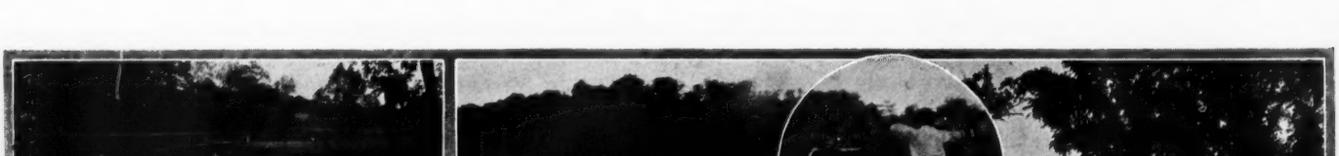
Experience with casings mounted on the Tire and Rim Association experimental drop center rim proved that the straight slope of the rim sides was unsatisfactory. Occasionally a segment of one bead or of both beads would not climb all the way out of the drop center portion, but freeze against the rim at some point below the line of the bead ledge.

The heretofore standard American 4.40 inner tube used with the flat base 3½ in. rim is both too long and sectionally too small to fit properly on the drop center rim. The proper fit of a tube on a drop center rim was worked out by utilizing a slotted rim, which permits of observing the seating of the tube at all points around the rim.

No flap is required in the drop center rim, but since a combination casing will fit both flat base and drop center rims, a flap must be provided and properly marked. The steel spoke, drop center wheel is fitted with a fabric-and-fiber rim strip that covers the base of the drop center portion and protects the tube against fouling on rims that have been long in service.

In summing up, Mr. Lemon expressed the view that the success of the steel spoke wheel with 4.40 in. drop center tire would largely determine whether or not the drop-center construction would be used for all sizes of automobile wheels in the near future. The familiarity of American motorists with the flat base rim is a factor against this change. If it be possible to develop a low-cost, lightweight demountable drop center rim, the solution might be found along that line.

The export situation is a factor in this connection. A growing proportion of the product of our automobile factories will go into foreign markets; export tastes must be carefully considered and cultivated, and these may have an influence on our wheel, tire and rim design.



Left to right: On the tennis courts; finish of the 50-yard dash which was won by Neil McMillan, Jr.; the men as well as the ladies enjoyed the archery; the trap shooters getting in their deadly work

An interested group watched the operation of the demonstrating model of a Dewandre vacuum brake

Anti-Freeze-

Radiator Liquids Analyzed to Determine Freezing Point, Specific Heat, Viscosity

J. F. Winchester, Standard Oil Co. of New Jersey, presided at the Anti-Freeze Session

A PAPER on "Anti-freeze Solutions and Compounds" was read by H. K. Cummings of the Bureau of Standards at the Thursday evening session, J. F. Winchester presiding. It was pointed out that lubricating oil has been used occasionally as a substitute for water in cold weather but the results may be unsatisfactory on account of the higher viscosity and the lower heat capacity of the oil as compared with water. The former necessitates wider passages and the latter a more rapid rate of circulation.

The chief requirements in a cooling fluid for winter use aside from a sufficiently low freezing temperature are that it must cause no damage to the cooling system through solvent action or corrosion, circulate freely at the lowest operating temperature, boil without decomposition near the boiling point of pure water, have a high heat capacity per unit volume, be readily obtainable at a reasonable price and be non-inflammable.

Kerosene has been used successfully although its heat capacity is less than one-half that of water. In mild weather its high boiling point may lead to overheating, and the odor and flammability of its vapor are disadvantages. It also has a somewhat destructive effect on rubber hose.

Compounds are Compared

A chart was given on which were plotted the percentage by weight of various compounds required to depress the freezing point of water to different degrees. The various compounds range as follows in order of effectiveness: Magnesium chloride, calcium chloride, sodium chloride (table salt), ethyl alcohol, ethylene glycol, glycerine, invert sugar, glucose, can sugar. To give protection down to a temperature of 22 deg. Fahr. requires 40.5 per cent of ethyl alcohol, 46 per cent of ethylene glycol and 56 per cent of glycerine, while only 26 per cent of calcium chloride would be required. While on a weight basis alcohol (up to 40 per cent) lowers the freezing point of water about twice as much as glycerine, equal volumes of denatured alcohol and 95 per cent glycerine are nearly equivalent in their effect.

Other properties besides the freezing point that were covered in the paper were the specific heat, which changes somewhat with the density of the solution, and the viscosity. Attention was paid particularly to alcohol, glycerine and glycol solutions but reference was made also to salt solutions. The corrosive actions of the different solutions were also dwelt upon. The conclusions were given in tabular form, the table giving certain obvious disadvantages of various types of radiator liquids. It also gave nine commercial materials, any one of which, if added to the water in the proportion specified, will absolutely prevent freezing down to 0 deg. Fahr. The specific gravity and boiling point of each solution was given, as well as its viscosity at 32 deg. Fahr. This table is produced below.

Table I.

Outstanding objections to various radiator liquids (for use in present day cars and trucks.)

Lubricating oils—High viscosity and low heat capacity.
Kerosene—Low heat capacity.
Denatured alcohol—Low boiling point.
Methanol—Low boiling point and toxicity.
Salt solutions—Electrolytic corrosion.
Sugar solutions—Viscosity.
Glycerine and Glycol solutions—High initial cost.

Quantity of Various Materials Required to Prevent Water From Freezing at 0° Fahr.

Material Used	Quantity per Gal. of water	Spec. Grav. of Solution at 60° Fahr.	Boiling Point of Solution Deg. Fahr.	Viscosity 32° Fahr. (Centip's)
Denatured alcohol (90% by volume)	2.5 qu.	0.958	187	6.9
Methanol (97% by volume)	1.6	0.986	185	4.2
Honey	8.5	1.296	225	78
Glycerine (C. P.) (95% by weight)	2.7	1.112	221	10
Radiator glycerine (60% by weight)	8.5	1.112	221	10
Ethylene glycol (95% by weight)	1.9	1.048	219	5.3
Common salt (100% Na Cl)	2.2	1.159	221.5	3.5
Magnesium Chloride (47% $Mg Cl_2$)	4.4	1.143	220.6	3.6
Calcium Chloride (75% $Ca Cl_2$)	3.0	1.178	220.8	3.1

In opening the discussion, G. A. Round of the Vacuum Oil Co. said his company had planned to market a petroleum product as a non-freezing cooling fluid and had instituted experiments to that end. It was found, how-

H. K. Cummings, Bureau of Standards, explained possible solutions for the anti-freeze problem

ever, that the low specific heat of all such products was a detrimental factor. All cooling systems are designed to do their work when using water, and if a liquid of materially lower specific heat is substituted for water and the rate of circulation is not increased, the engine is likely to heat up during periods of moderate temperature. He had found that with glycol it was necessary to look out for leaks, the glycol solution having the effect of reducing the thickness of gaskets and hose which necessitates taking up on cylinder head bolts and hose clamps.

Byrd's North Pole Experience

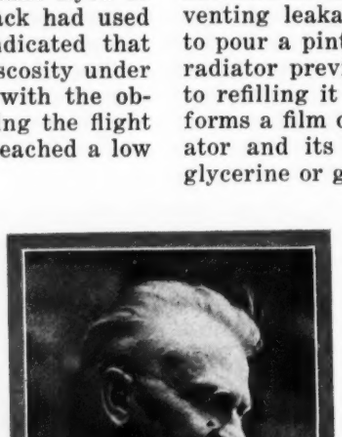
One member pointed out that the fact that Byrd in his recent flight to the North Pole and back had used glycol solution for his cooling system indicated that there need be no trouble from excessive viscosity under low temperature conditions, but this met with the objection that Byrd did not make a stop during the flight and that the cooling fluid therefore never reached a low temperature.

Charles L. Sheppy said the impression prevailed that glycerine solutions when heated up formed a deposit which was likely to clog the radiator passages. To investigate this he had exposed such a solution to a high temperature for a long time but had found no deposit. In this connection the author of the paper stated that the first effect of glycerine when introduced into the system is somewhat opposite to this, in that it tends to open up the seams.

J. F. Winchester said in the operation of large fleets the cost of the non-freezing fluid is an item of some importance. To determine the relative costs he had experimented with three fluids, alcohol, glycerine and glycol solutions. Although alcohol cost only 51 cents a gallon while glycerine cost $3 per gallon and glycol about the same, the latter two solutions had advantages in certain sections of the country for the reason that the solution need not be renewed.

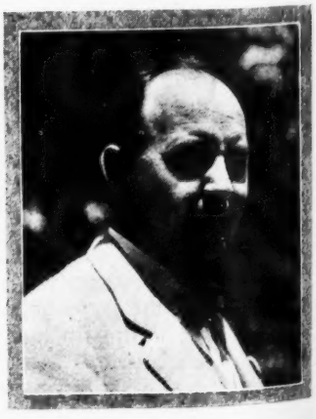
Trouble from evaporation of the alcohol is most likely in Southern cities, where the temperature at certain seasons of the year rises quite high during the day-time, causing the alcohol to be driven off, while the nights are cool, and may then cause freezing and bursting of the cylinder jackets. In the colder climates the alcohol solution would probably be the cheapest.

The use of the more viscous non-freezing fluids was said to be generally impractical with thermo-siphon cooling systems, as the rate of circulation will be so much reduced. H. F. Blanchard said he had poured a can of radiator liquid in with the glycerine solution and had had no trouble from leakage. Another way of preventing leakage under such conditions was said to be to pour a pint of cylinder oil on top of the water in the radiator previous to draining the radiator with a view to refilling it with non-freezing solution. This oil then forms a film over the whole interior surface of the radiator and its connections which seems to prevent the glycerine or glycol from penetrating the joints.



Right: Andrew Bachle, vice-president, engineering division, Paige-Detroit Motor Car Co.

Charles S. Crawford (below), chief engineer, Stutz Motor Car Co. of America, was present and took part in the discussion at the Gear Session



The group below, interested in springs and bumpers, got together during the meeting to discuss their engineering and manufacturing problems

Indicating High-Speed Engines

H. M. Jacklin demonstrates method of using ordinary gas engine indicator by means of an adapter.

PROF. H. M. JACKLIN, of Ohio State University, as a feature of the meeting, demonstrated a method of using an ordinary gas engine indicator for indicating high speed engines, by means of an adapter invented by himself and C. P. Roberts. A number of diagrams taken with this instrument on a single cylinder Delco-Light air-cooled engine were displayed.

The adapter is so mounted on the engine that the manifold (which is used when indicating multi-cylinder engines) is located near the middle of the cylinder head lengthwise, and either on the same level or higher. The manifold connects by air-or-water-cooled tubes to the compression cock openings of the engine.

The indicating device comprises a housing of upright form from which both ends of a horizontal 5/8 in. shaft project near the bottom. This shaft is placed in driving connection with the accessories shaft so as to be driven at crankshaft speed.

A slow-speed indicator is connected to this device at *A* (see cut) and is arranged with its pencil motion pointing downward, since its cylinder must be filled with rather heavy lubricating oil. The oil materially improves the seal of the indicator piston and also reduces the clearance in the cylinder, hence the amount of gas flowing from the cylinder indicated to the indicator cylinder, or vice-versa, when the effective pressure varies, is reduced.

Drum Oscillates at Low Speed

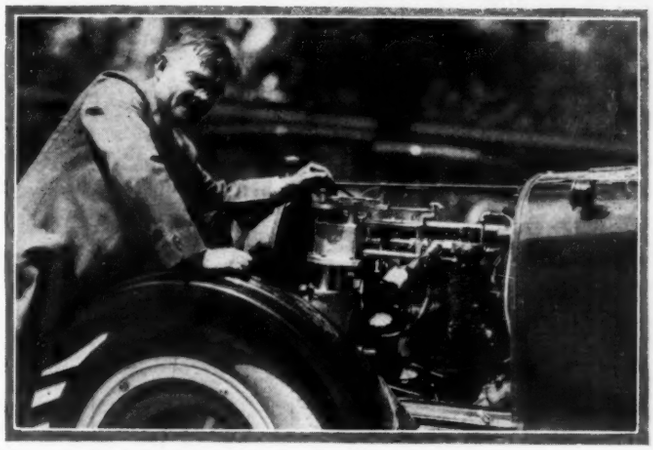
The drum of the indicator is oscillated through a string from a crosshead incorporated in the indicator device, at a very slow speed as compared with the speed of the engine piston, the crosshead making only one stroke to 800 strokes of an engine piston. The reason for using so low a drum speed is that it eliminates all trouble from inertia effects.

A connecting rod and crank mechanism controls the crosshead. The connecting rod is made adjustable as to length, so the crank-connecting rod train may be given the same characteristics as that of the engine being tested. The throw of the crank is 1 5/8 in. The cross-head guide may be adjusted to simulate any offset of the cylinder in an engine under test.

A shaft extending vertically upward from the 5/8 in. shaft is operated at one-half engine speed. There is another horizontal shaft carrying a hand crank, which is driven (when a clutch is engaged) at 1/40 engine crankshaft speed. This horizontal shaft drives the vertical shaft to which the graduated crank disk is attached, through worm gears having a reduction of 20 to 1. Therefore, with the clutch engaged, the graduated crank disk is driven by the engine at 1/800 of engine speed. With the clutch engaged, the crank disk may be turned in either direction by the operator by means of a hand crank.

A small poppet valve is arranged inside of part *A* which is opened for a very small interval of each cycle of the engine being tested. The valve head is located to the left

(Continued on page 1022)



Prof. H. M. Jacklin of Ohio State University with his high-speed engine indicator mounted on a car

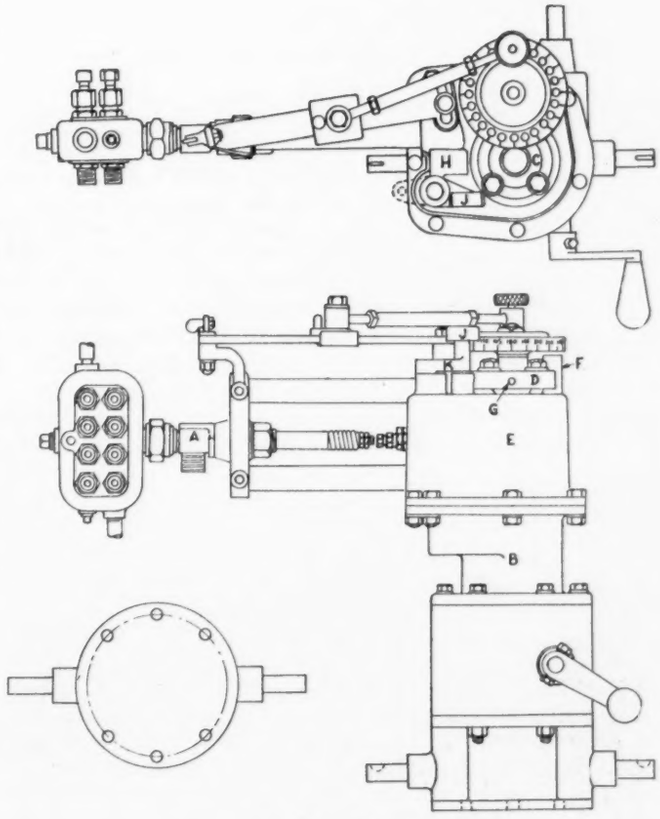


Fig. 1. *Two views of J. R. indicator adapter*

Just Among Ourselves at

SECTION stunts were frequent and snappy. The Dayton Section provided the most spectacular stunt of the meeting when it brought flyers from McCook field to photograph from the air several hundred members and guests lined up on the golf course in such a way as to form the letter S. A. E. The photographs were developed in the air and copies were available within a few minutes after the event took place.

* * *

BUT for forceful presentation the Detroit High Hats, lead by Phil Overman and his ukelele, with Clayt Hill at a peripatetic pipe organ, won the Rayon golf club.

The airplane which came from McCook Field to do its aerial photography stuff created a lot of interest among the natives. The picture stunt, arranged by the Dayton Section, was a huge success

With tall red hats adorning their none too high brows, the daily dozen troubadours and their portable organist rolled in and out of the dining room warbling wails that left no heart untouched by their sensuous bathos—whatever that is. Trying to describe further would be sure to mean ending up with—"Well, anyways it was awfully funny when I heard it," so we won't try. Only must add that their ticket prize dance at the ball on Thursday made happy the hearts of the ladies.

* * *

F. E. MOSKOVICS, having shed the war paint with which he led the Indianapolis braves at White Sulphur last year, appeared on Thursday evening in the role of Earl Carroll and presided over the bathtub scene put on with stark realism by the Indianapolis Section, under the title, "Pluto Vanities." It was predicted that this burlesque would be a wow. It was even better. It was a wow wow—if you get what I mean.

* * *

THE Daily SAE! We always are accustomed to pay a polite tribute to the Daily SAE and often we just let it go at that. But something special is in order this year. Hyatt Roller Bearing Co. was responsible for the sheet at French Lick and the job done was a knockout. We had a real newspaper—interesting, intelligent, funny, entertaining, lively. The French Lick Daily SAE was our idea of a hard task admirably done—and that's no apple sauce either.

* * *

TO the Easterners, French Lick seemed like the end of the earth before the completion of the long train ride necessary to get there. The chief occupation on the way out, besides playing bridge and swapping lies, was setting back watches. When other conversation ran out a good argument as to what time it was could always be started. That never ran out. But that ride was forgotten immediately upon arrival. Everybody got a room, got it quickly and efficiently and didn't have to fight for it. It was a great place for a meeting and everything ran like a well-oiled watch—after they got the members at large chosen for the nominating committee. The good old constitution finally ironed out that confusion which had the business session tied up in knots for fifteen minutes, however, and a good time was had by all, including the Scandinavian.

* * *

HIGH speed indicators are delicate devices, and there are difficulties enough connected with their use on engines on the block to have discouraged any one from using them on a moving vehicle. By a device invented by Prof. Jacklin of the University of Ohio an ordinary low-speed gas engine indicator can be made use of for indicating high speed engines, and as both the indicator of this type and the adapter used with it are quite robust, there is no reason why they cannot be used on an engine in service on the road. The indicator was demonstrated on a moving vehicle at the French Lick meeting and diagrams were produced which were of the large scale characteristic of the usual steam engine indicator diagram. The diagram is not a trace of pressure variations in the cylinder during a single cycle but is built up of what may be called elements of a large number of succeeding cycles. A rather interesting feature of

The instrumental portion of the Detroit High Hats with Clayt Hill at the organ

the S. A. E. Summer Meeting

The "Apple Pie" golf course warmed the hearts of the amateurs. A committee of fourteen spent the entire four days looking for the rough

these diagrams is that, whereas they indicate practically no variations from cycle to cycle in the gas pressures during most parts of the cycle, considerable variations are indicated in the pressures during the early part of the power stroke, corresponding to the inflammation period of the gases. This is evidently due to variations in the character of the mixture which happens to be in the vicinity of the spark points at the moment of ignition, and consequent variations in the speed of flame propagation.

* * *

SPORTS at the S. A. E. summer meeting now really mean Golf with a big G. The golfers outnumber the other sport addicts about fifteen to one, only a few enthusiasts keeping alive the competitive features of many of the other events. But those enthusiasts make up in activity what they lack in numbers. Particularly deserving of mention is the way in which Bert Lemon steamed up interest and activity in the tennis tournament despite the gopher holes which dotted the courts and the somewhat faulty mathematics of the lining. Incidentally, he was runner-up in singles in the tennis tournament.

* * *

A LOT of the boys came away from French Lick with better golf scores for 18 holes than they ever had before. When they talk about their achievements don't forget to ask them on which course they did their stuff. Immediately adjoining the hotel was an apple-pie course which made the heart of the dub leap up within him and rejoice and be exceedingly glad. A committee of fourteen spent the entire four days looking for the rough on this course, while the only obstacles to progress visible were a few trees scattered here and there and the natural incapacity of the human being to strike a small round sphere squarely with a long queerly shaped stick.

* * *

BUT the Hill course, where the real golf was played, made up for the easy going one might enjoy on the apple pie course. We were informed that the Hill course has on it 98 traps, but from personal investigation we can state authoritatively that this is incorrect. Ye editor personally knocked balls in and out of 1008 sand traps by actual count during the course of playing 18 holes. After that he lost count. But, of course, a golfer wouldn't find as many as a trained editorial investigator.

* * *

THOSE who drove to the meeting by automobile encountered conditions just the opposite from those met with in reaching White Sulphur last year. Instead of the almost impassable roads which greeted some of the drivers going to the sessions last year, fine straight highways made riding to and from the gathering a real

The Hill course, where the real golf was played. It is reported that many who started out over this course are still floundering among the 98 traps

pleasure. Some of the roads were rather dusty, but all were smooth and in good condition, making possible high average speed for the convention travelers.

* * *

LUKE BARNETT, the vaudeville head-waiter, made things warm for the incoming diners on Thursday evening. He officiated officiously, made men wilder and worked himself up to supreme indignation before announcing that after all it was only a joke. This popular entertainer, who already has done his stuff before such automotive audiences as the N.A.D.A. and the Paige-Jewett dealer dinner in Chicago, was brought to French Lick as a Cleveland Section stunt.

* * *

ROLLIN ABELL, of single sleeve valve engine fame, was thought by some at the meeting to have been responsible for the Abell Special which paticipated in the Indianapolis Race this year. This was not the case as neither Rollin Abell nor his engine had any connection with the racing job. The Abell Special was a rebuilt Miller job entered by a Californian and driven by Houser.

* * *

EVERYBODY seemed to have a good time and keep well despite the all-night sessions staged by several groups and despite the power of suggestion involved in the name of one of the house doctors whose cognomen was O. Sicks.

Indicating High-Speed Engines

(Continued from page 1019)

of *A*, so that when opened it establishes communication between the manifold and the indicator attached to *A*.

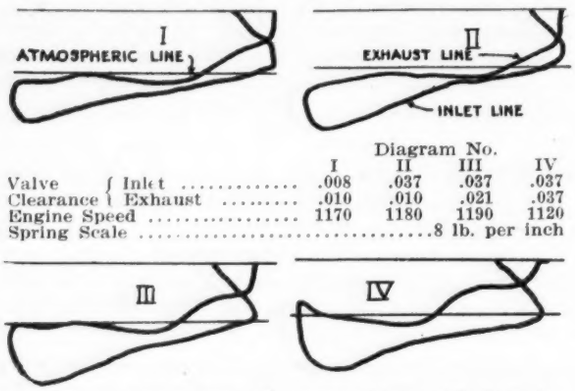


		Diagram No. I	II	III	IV
Valve Clearance	Inlet	.008	.037	.037	.037
	Exhaust	.010	.010	.021	.037
Engine Speed		1170	1180	1190	1120
Spring Scale		.8 lb. per inch			

Fig. 2. *Lower loop diagrams taken with J. R. indicator showing effect of valve clearance*

The vertical shaft is in two parts, the upper being driven from the lower through a conventional differential mechanism. To the upper shaft is attached the small disk *C*, the purpose of which will be explained later. Disk *D* has a hollow extension which encloses the upper shaft and which carries a cam within the housing at *E*. This cam operates the small poppet valve through a suitable rocker lever which is also enclosed in the housing and an adjustable tappet, the end of which is shown.

The movement of the differential carrier and of the satellite gears is controlled by suitable gears between the crank disk shaft and the differential carrier. By moving the differential carrier, disks *C* and *D*, when clamped together as shown, may be moved faster or slower than the lower vertical shaft. This means that the cam may be advanced or retarded in reference to the engine cycle. The gears between the crank disk shaft and the differential carrier are so proportioned that the cam is advanced or retarded one full turn for two turns of the crank disk. Since the crank disk shaft rotates very slowly, the cam is moved only very slightly from cycle to cycle.

Thus, the pressure within the engine cylinder forces a small amount of gas into or from the indicator cylinder at a certain point in the cycle. While the engine crankshaft makes the next two turns, the crank disk moves a very small distance which moves the indicator drum through the connecting rod, cross-head, and the cord; and the cam is moved slightly forward or backward so that the valve is opened in the next cycle at a slightly different point. Thus, only a small element of the diagram is obtained from an individual cycle.

The crank disk is made with 24 drilled and reamed holes, spaced equally on a circle of 1 5/6 in. radius. The stroke of the cross-head is then 3¼ in. A movable crank arm on this crank disk is held in position by the point of a knurled screw fitting into any hole in the crank disk, and for indicating the different cylinders of a multi-cylinder engine the crank arm is moved through angles equal to those between the crank throws (180 deg. in a four-cylinder engine, for instance). This standard disk may be used on any engine where the firing intervals are in multiples of 15 deg.

Offset diagrams are made, in the case of a four-cylinder engine. by using hole 270 for cylinders Nos. 1 and 4 and hole 90 for cylinders Nos. 2 and 3. In like manner a hole at 90 deg. to the hole used to obtain a power diagram can be used in all cases for obtaining offset diagrams.

In obtaining lower loop diagrams, the same holes are used as for the power diagrams, but a rather light spring is inserted in the indicator and a stop provided on the piston to protect the indicator spring against damage.

Professor Jacklin considers that there is a distinct advantage in a composite diagram of a great number of cycles, over any other kind. The diagram is built up before the eyes of the test engineer so that he can always have a check on operating conditions from the engine end of the test equipment. The diagram is then available for study and analysis as soon as drawn, without recourse to photography.

By means of the crank disk, any part of the cycle, such as the combustion line, may be extended over a great part of the diagram thus enabling a more minute study to be made. Offset diagrams, as stated before, may easily be made to show the valve action in greater detail than the lower loop diagrams reproduced herewith.

Rapid and accurate work can be done with this apparatus in investigating the fluid processes in multi-cylinder engines. Variations in valve timing, valve lift, ignition timing, and other things affecting individual cylinders become immediately apparent and may be corrected.

Thermal Efficiency Calculated by Dr. Treves Probably Unobtainable with Ordinary Fuels

(Continued from page 981)

this condition—equal compression pressure—the calculated thermal efficiency should be reduced in the ratio 0.4805 to 0.413.

The other assupmtion made, and which does not seem justified under any conditions, is that when an engine is supercharged the proportion of fuel mixed with a unit weight of air will vary inversely as the square root of the density of the air but that the energy of a unit weight of combustible charge is not affected by this "leaning."

It is true that with a simple carburetor without corrective devices the richness of "quality" of the mixture is affected by the density of the air at the carburetor inlet. This relationship is particularly apparent in the case of aircraft engines used for altitude flying. With increasing altitude, as the density of the air at the carburetor inlet decreases, the mixture becomes richer and richer and a corrective must be applied by means of a so-called altitude control. Hence, if the density of the air at the carburetor inlet is increased, by means of a supercharger, the mixture will tend to become leaner, the amount of fuel in a unit weight of air varying inversely as the square root of the density of the air.

But we can hardly expect to make use of this relationship to improve our thermal efficiency because without supercharging we can lean our mixture down to the most economical proportion, and if supercharging leans it down more it will be necessary to apply a corrective. Therefore instead of figuring on a reduction in fuel consumption in the ratio of 1.265 to 1 when supercharging to 1.6 atmospheres, we cannot expect any reduction at all from this source.

Therefore, if an engine were supercharged to 1.6 atmospheres and the maximum compression and the quality of the fuel mixture remained the same the thermal efficiency would be reduced in the ratio $\frac{0.413 \times 1}{0.4805 \times 1.265} = \frac{68}{100}$, that is, to 68 per cent of the thermal efficiency without supercharging.

This, of course, would not condemn supercharging, for if the supercharger were used only when more than the maximum power of the engine operating without supercharger was needed and the engine were reduced in size, its average economy might still be greater than that of a larger engine without supercharger, because of the higher power factor of the smaller engine.—*Editor.*

THE FORUM

Ricardo Thinks Short-Stroke Engine Has Several Advantages

Correspondent quotes British engineer in support of argument that a long stroke does not necessarily make a better engine.

Editor AUTOMOTIVE INDUSTRIES:

Fred Thomer, in his letter of April 22, is bringing forward several arguments in favor of the long stroke engine, and complains that at the present time but little attention is being paid to the matter of stroke bore ratio. It appears to me that not all the experts agree with this statement, at least Harry R. Ricardo seems to be of another opinion. I should like to draw the attention to an article in "The Automobile Engineer," of July, 1925, wherein Mr Ricardo is giving quite another viewpoint of the matter of stroke-bore ratio.

Quoting Mr. Ricardo: "The principal argument, and it certainly is rather a strong one, in favor of the short-stroke engine, is its great rigidity, particularly in respect of the crankshaft, which makes for sweeter and quieter running.

"In actual practice, unless there be some artificial limitation, such as the taxation on motor car engines in England, which confers a marked advantage on the long stroke engine, the stroke-bore ratio is really governed entirely by questions of mechanical expediency. Just as efficient an engine can be made with the stroke double the bore or even considerably less than the bore, if the designer is free to put the valves where he likes, and for that matter equally as efficient an engine can be made with the valves at the side, or in the head in a single row, if the stroke-bore ratio can be made whatever is most desirable.

"In the case of the side-valve engine, the efficiency increases slightly as the stroke-bore ratio increases, and the higher the compression, the more does it increase, for the simple reason that with a short stroke and a high compression there is not room to provide sufficient depth of combustion chamber and still leave adequate space around the valves."

Mr. Thomer, making a comparison between an engine of 4⅝ in. bore by 6 in. stroke, and one with 4 in bore by 7½ in. stroke, comes to the conclusion that the latter shows a decrease in inertia forces of 25 per cent. It is not quite clear to me how Mr. Thomer arrives at this conclusion. Assuming both engines to run at the same number of revolutions and having a reciprocating weight of pistons and connecting rods proportional with the area of the piston, the long stroke engine only shows a decrease in inertia forces around 6 per cent.

Due to the longer stroke the acceleration becomes larger in proportion to the increase in stroke, while the reciprocating mass becomes smaller proportional with the area of the piston, giving a resultant decrease of inertia forces as mentioned.

Mr. Thomer also made the following statement: "And last but not least, there is the consideration of overall length, which again turns out in favor of the long stroke, since the distance between cylinder centers is determined by the bore." I do not believe Mr. Thomer to be quite correct in this. The question of valve size is governed by the piston displacement and number of revolutions rather than by the bore of the engine, and an efficient layout of the valves is rather what determines the distance between cylinder centers than anything else, or at least limits the bore-stroke ratio.

Also, the question of a compact combustion chamber and short flame travel is more a question of efficient valve layout, at least with a high turbulence type combustion chamber, which is the more sensible step to take when striving for high efficiency.

The stroke-bore ratio is hardly of any influence here, because that part of the head which covers the biggest part of the cylinder bore (only allowing a very small clearance for the piston at the end of the stroke) is not to be considered as being part of the combustion chamber. The slightly larger cooling area with the short stroke engine has hardly any effect on the efficiency of the engine, as also was pointed out by Mr. Ricardo: "The loss of heat to the cylinder walls has nothing like the influence generally attributed to it, and in the case of a well-designed engine the complete suppression of all heat losses to the cylinder walls would increase the power and efficiency by about 7 per cent."

It is a well known fact that a smaller cylinder allows a higher compression ratio, but this is not a question of bore-stroke ratio.

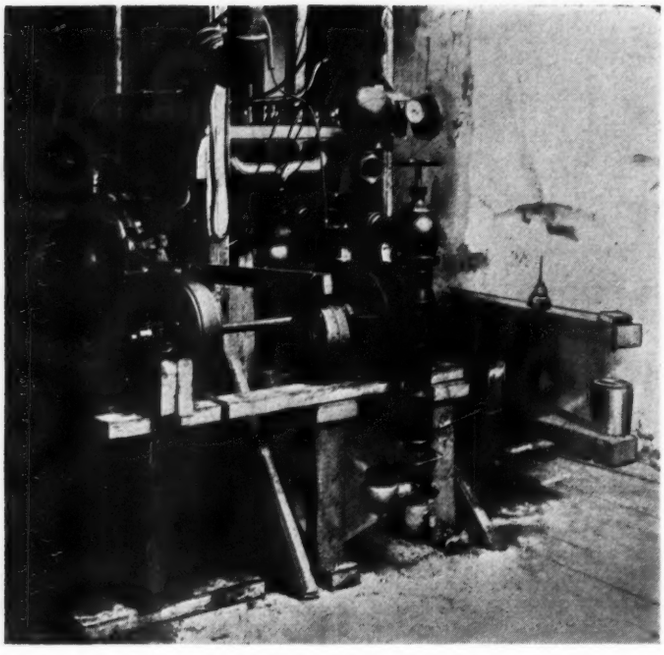
CHRIS. H. BOUVY.

Early Use of Superchargers

Editor AUTOMOTIVE INDUSTRIES:

Relative to the discussion as to when superchargers were first used in the United States, the enclosed photo shows a four-cylinder automobile engine belted to a Connersville blower, rigged up and tested by the writer in 1911 on South Michigan Ave., Chicago.

The blower, which I believe, was the smallest Connersville type we could get, ran at the same speed as the engine and was so connected that gas from the carburetor to the engine could pass through the blower or around it, depending on the opening of the angle valves shown in photo. A Prony brake absorbed the load, but on account of the engine's great speed with the supercharger working, and the



Connersville blower as rigged up with automobile engine for tests in Chicago in 1911

excessive vibration, standard scales could not be used for weighing the load; therefore a known weight, which could not be raised without the blower, was attached to the brake lever and when speed increased the weight would be raised, showing that the horse power increased faster than the speed.

Above normal engine speed the vibration was so great that the engine, which was tested on the second floor of the building was anchored to both floor and ceiling, as shown by the vertical struts.

C. E. SARGENT.

Goes Back to 1907

Editor AUTOMOTIVE INDUSTRIES:

In "The Forum" April 22, issue of *Automotive Industries*, I note that Mr. Short would lead us to believe that superchargers were first used in this country for racing purposes in 1913.

Superchargers were a rather old story in the racing engine game in this country long before 1913 and for the information of Mr. Short, his colaborators and critics I would refer you to paragraphs a-b-c and d of Section 4, Rule 3, beginning on page 15 of the 1907 Racing Rules Book of the American Power Boat Association.

The above mentioned rules prove conclusively that superchargers were in common use on racing engines even before 1907.

OLIVER E. BARTHEL.

Superchargers in 1912

Editor AUTOMOTIVE INDUSTRIES:

In the March 25, 1920, issue of *Automotive Industries*, in an article by S. Gerster and W. F. Bradley on the designing of 183-cu. in. racing engines, there is a full page sectional drawing of the French Sizaire-Naudin racer built in 1912, probably for the Grand Prix race. This engine was equipped with a supercharger which seems to be much like those used on the present-day racers.

The supercharger was evidently of the centrifugal type and was driven by a chain at the rear end of the crankshaft at several times crankshaft speed. It seems to have been placed ahead of the carburetor. It is stated in the article, however, that the spark plugs would stand up only a few minutes, so it could not compete in the race.

The same year the Hispano-Suiza firm entered a car having a supercharger. In this car a two-cylinder air compressor was used. This car also had difficulties and did not start.

It would seem then that Europe was ahead of us in the use of the supercharger in racing, if the Chadwick racer mentioned by Mr. Short in The Forum recently did not come out until 1913. However, if the Chadwick machine actually competed in a race, it deserves as much credit as the two others mentioned.

DONOVAN E. SHUMARD.

Studebaker Beacon Wins

Editor AUTOMOTIVE INDUSTRIES:

In the March 18 issue of *Automotive Industries* you printed a letter written by Mr. Rigby of the Studebaker Corporation, in which he claims that the credit for installing the first aviator's beacon should go to the Studebaker Corporation. As far as New York University's beacon is concerned this statement is correct, inasmuch as the Studebaker beacon apparently was put into operation about a year and a half before the one at the University. The beacon at the University was first lighted on May 26, 1925.

When I wrote before I certainly had no intention of starting a contest to see who had the first beacon. I did not claim that New York University had the first beacon. I will admit, however, that the average reader might be led to interpret my statements in that way.

ERWIN H. HAMILTON,
Asst. Professor of Automotive Engineering,
New York University

Road Congress in Italy

AN international road exhibition is to be held in connection with the samples fair which opens at Milan, Italy, on September 6, next, and at the same time there is to be held an international road congress which, while being opened at Milan, will end at Rome. All nations have been invited by the Italian Ministry of Foreign Affairs to participate in both congress and exhibition. The King of Italy has consented to become patron of the congress, and Signor Mussolini is to be chairman of the committee of honor. The congress is being organized by a special commission with headquarters at Milan in the offices of the State Civil Engineers. This committee acts as the local organ of the International Road Association, the headquarters of which are at Paris. The previous analogous congresses were held in Paris and in London before the war, and at Seville after the war.

The congress will deal chiefly with the following—(*a*) Concrete Roads; (*b*) Bituminous and Asphalt Surfaces; (*c*) Standardization of Tests for Coal Tar, Bitumens and Asphalts used in Road Construction; (*d*) Census of Traffic; (*e*) Development and Planning of Towns in Relation to Traffic Interests; and (*f*) Automobile Roads. The congress will probably assemble in the Castello Sforzesco and hold its sittings in the large building of the Facolta di Giurisprudenza, which is not far from the Sample Fair.

EDITORIAL

Extending Heat Treatment Studies

A DECISION taken at a recent meeting of the American Society for Steel Treating, to go more actively into the subject of non-ferrous metals and alloys, may prove of considerable importance to the automotive industry. There is no reason why the Society should not widen the scope of its activities, even though its name would seem to limit it to the field of ferrous metals. Up to a few years ago the only commercial metals that were responsive to heat treatment were the ferrous metals, but we now have duralumin as a leading example of non-ferrous alloys which can be improved greatly by such treatment.

There is little doubt that many failures in the use of non-ferrous alloys in the past, and particularly of aluminum alloys, have been due to lack of information regarding the requirements of these alloys in fabrication. It is now pretty well agreed that all parts made of light alloys which in service are subjected to high temperatures, must be annealed at suitable temperatures after being machined to size. There is much opportunity also for research work on other non-ferrous metals that are coming into use, such as magnesium. Probably no class of men are better fitted to take up this line of work than those making up the membership of the Society for Steel Treating, and the decision of the Society to expand into the non-ferrous field is to be commended.

"Selling" the Street Car

WITH the constantly increasing popularity of the automobile and the motor bus making them more worried every year about future business, the electric railways in many cities are conducting active campaigns to sell the public on street car transportation.

Positive, constructive sales effort on the part of the electric lines is not only natural, but entirely warranted under the circumstances. But when these efforts are tied up with an attempt to decrease the sale of automobiles they may fairly be criticised.

Indianapolis street cars, for example, are carrying a sign, prominently displayed, which reads: "Why buy the car when you can buy a ride at a time on the street car?" The automotive industry has never gone out to suggest that people stop riding street cars, however strongly it has merchandised the desirability of its own products. Positive selling almost always turns out to be more profitable than negative selling.

A good example of the positive brand of selling as applied to street railway transportation is to be found in Grand Rapids, where the railways are now attempting to get what they consider to be their just share of the business by placing at the disposal of the public a new type of street car which is designed as far as possible along automotive lines. The cars are advertised as "electric railway coaches," are finished in striking colors, have a skirt around the bottom to give a low-hung appearance, are fitted with vibration-dampened trucks and wheels and silent gears to reduce noise, and the seats are of the individual chair type, arranged in pairs, to further increase the comfort of the passengers.

This is constructive merchandising. If the street railways expect to maintain their place in the transportation picture they must proceed along the line of improving the general character of their equipment.

Racing and Design

THERE is one school of thought in the industry which believes racing to be of little real value from the standpoint of influencing design and looks on the races as purely sporting events. Another group feels strongly that racing is a melting pot of design ideas; that it is an important experimental laboratory.

Whichever group may be most nearly correct in its conception, the race session at the Society of Automotive Engineers' summer meeting proved one thing clearly: automotive engineers in general are interested in the technical developments brought about in the racing cars, they are interested in hearing what the racing enthusiasts have deduced from the performance of their speedsters, and they are following fairly closely the developments which are going on in the racing field.

The race session was the liveliest of the sessions at the meeting. The picturesque sport behind it generated an enthusiasm for listening to the information presented which was not and rarely can be developed for listening to presentation of results brought out of the formal laboratory. The latter results probably are more economically obtained and more important *per se*, but they often do not get the same rapid attention—and hence the same rapid application—that frequently is accorded design ideas developed in racing.

The Business Situation

"WHILE there is reason for thinking that commodity prices and production will continue to recede until readjustment is completed, there is no indication that prices will fall to a depression level. Speculation in commodities has been notable for its absence. No signs of a general glut in commodity markets are visible and, therefore, no necessity for protracted liquidation exists."

The foregoing is offered by the First Wisconsin National Bank as its view of the current economic situation. It is a reasonable and probably accurate view of the present status of business.

AUTOMOTIVE NEWS SECTION INDUSTRIES
Philadelphia, Pennsylvania Thursday, June 10, 1926

Continued High Retail Sales Favors Firm Production Rate

PHILADELPHIA, June 10—With motor car and truck production reaching an estimated total of 420,000 to 430,000 vehicles in May, against 455,000 in April the industry continues in one of the most amazing periods of its history. The record activity of the year to date has overthrown the calculations of the most enthusiastic observers of trade conditions, and meanwhile sales are continuing at a level that makes any drastic cuts in output unlikely to occur for several weeks.

Although June production will probably be somewhat under May, this will be due as much to plant adjustments in preparation for new models as to any other factors. The unexampled optimism with which some of the leading companies in the industry are looking to the future was shown in the week just past by the announcement of further plans for enlargement of factory capacity.

Probably no one believes that the second half of the year will equal in volume or profits the first half, and the producing companies are preparing the way for a successful negotiation of a period of harder sales. The development of more economical production processes, together with the designing of new cars expected to catch the public approval and even to tap new buying markets, has been proceeding rapidly in the last few months, and a slump in sales will be quickly met with measures well calculated to rejuvenate the market.

Meanwhile the industry is showing the greatest interest in the next move of Henry Ford. The Ford company has lost ground this year to other large factors in the business and everyone expects him shortly to take some drastic action. The Dearborn and Highland Park factories in the last two months have undergone sweeping reorganization of personnel apparently with the object of promoting efficiency and lowering costs, but whether this presages improved models or price reductions or both is an element of mystery.

To some extent, of course, the plans of producers of cars near the Ford price level depend upon the action taken by the Ford company.

Nash May Sales Increase

KENOSHA, June 10—Shipments of Nash Motors Co. to its dealer organization in the nine months since Sept. 1, 1925, totaled 101,402, reports E. H. McCarty, general sales manager. Shipments in the first five months this year have exceeded shipments for the first eight and one-half months in 1925. May sales showed a gain of 44.3 per cent over May last year.

WILLYS ANNOUNCES LIGHT CAR COMING

TOLEDO, June 10—Willys-Overland Co. announces that it shortly will bring out a new light 4-cylinder car which will sell in the low-priced field and which it expects will become a leading contender. A general announcement of the details of the car will be made about July 1.

To prepare the way for the new car which will be of the small car type of construction and will be capable of high speeds with low gasoline consumption, the company during the past several months has released a series of publicity stories dealing with the adaptability of cars following European design for the American market.

The Overland company has pointed out in its advertising and publicity the advantages of a short, low hung chassis with small wheels and four-wheel brakes, and powered with a high speed engine of superior efficiency, so it seems logical to conclude that the new car will embody these features of design.

Continental Ready to Produce Argyll

DETROIT, June 10—Continental Motors Corp. reports an increase in surplus of $1,011,761 for the six months ending April 30, and profits from operations for the period as $2,016,441 before depreciation interest charges and taxes. After deductions $1,011,578 is available for dividends. Cash on hand and in banks is listed at $4,065,150. Inventories amounted to $7,142,466, and accounts receivable $2,137,194. The ratio of current assets to current liabilities is 5.7 to 1. The statement shows a surplus account of $10,656,036.51.

"The expense of experimental work on the new Continental Argyll single sleeve valve engine has been taken care of during the period and your company is now ready to proceed with production of this latest development in internal combustion engines," declared R. W. Judson, president.

"The growth of the division of industrial engineering during the last six months has been steady and consistent. New customers have been obtained and orders are being received in increasing numbers. Schedules from customers are being satisfactorily maintained and new business has been closed with several large motor car manufacturers. Never before in the history of the company have the possibilities been as favorable as they are now and the outlook for the last half of the fiscal year is exceptionally favorable."

Citroen Extends Capital to 125,000,000 Francs

PARIS, May 29 (*by mail*)—By full page advertisements in all the Paris dailies, Citroen announces the issue of 500,000 ten year bonds of a nominal value of 500 francs, bearing 7½ per cent interest, with, in addition, 1 per cent of the firm's turnover to be distributed annually among holders. The total issue is therefore for 25,000,000 francs. It is claimed that on the basis of present sales the return on these bonds would be 20 francs per year, plus 37 frs. 50 c. interest.

The Citroen company's present capital is 100,000,000 francs, all of which has been privately subscribed, with control vested almost entirely in the hands of Andre Citroen. It is understood that periodically, at certain periods of the year, Citroen has been handicapped by reason of lack of capital, but up to the present has preferred to rely on bank loans rather than to appeal to the public for additional capital.

National Truck Users Meet

NEW YORK, June 8—An important meeting of the National Truck Users Conference will be held at the City Club here next Monday at 1 p. m. This conference was formed to present the views of truck operators at the hearings on the Cummins bill for Federal regulation of truck and bus common carriers.

Make New Oilless Material

PARIS, May 12 (*by mail*)—Under the title "Silentbloc" the Repusseau Company of Paris, a firm holding the European concession for Hartford shock absorbers, has produced a new oilless bushing material, which is being made use of for spring shackle bolts, shock absorber fittings and other similar parts. It is the invention of a Belgian engineer.

U. S. Court Clears Truck Regulation

Distinction Between Private and Common Carriers Seen Defining Future Laws

NEW YORK, June 9—The United States Supreme Court decision that states have no right to control through commissions the operations of automobile, bus and truck transportation conducted under private contract is of considerable importance, especially with reference to any law Congress may pass later for the regulation of such transportation.

The decision upholds the policy of the Motor Vehicle Conference Committee, which distinguishes definitely between common carrier and private truck service. Under the California law, which was upheld by the State courts and is now declared unconstitutional by the United States Supreme Court, private contract carriers were required to obtain certificates of public convenience and to submit to regulation by the State commission. Some of the California private contract carriers favored this law and others opposed it.

The U. S. Supreme Court now decides that states cannot compel a private carrier to assume against his will, by taking out a permit or franchise, the duties of a common carrier. The majority held that any law permitting the State to require a common carrier to surrender any of his rights, although the action might be voluntary, is compulsory and unconstitutional, and that no State can impose conditions upon a private carrier which will require him to surrender any of the rights of private contract which are protected by the constitution even though such surrender of rights may appear voluntarily given.

Three justices dissented from the opinion, insisting that states have the right to regulate the use of highways by persons operating under private contracts as well as those who announce themselves as public or common carriers.

Although the majority decision seems to be approved by bus and truck interests generally, the fact that there was also a minority opinion not generally approved by those interests is regarded here as indicating that these branches of the industry will continue to be confronted with this question as the whole subject of State and Federal bus and truck regulations develops.

Heil Company Completes Half-Million Expansion

MILWAUKEE, June 8—The Heil Co., manufacturer of steel dump body and tank equipment for motor trucks, and maker of the Hydro-Hoist, has started construction work on a new machine shop, 100 x 300 ft., to be used mainly for the production of the hydraulic hoist. The new shop is the final unit of plant extensions costing about $500,000 which have been in progress for the past year and are being completed at the time the Heil Co. is observing its silver jubilee. Units already completed embrace a template shop, addition to the power plant, a garage, a new administration building and various shop extensions. Julius P. Heil is general manager.

Edmunds-Hall Corp. Waits Stock Action

DETROIT, June 8—Directors of the Edmunds & Jones Corp. and the Hall (C. M.) Lamp Co. will submit plans to stockholders of the two companies for a consolidation under the name Edmunds-Hall Corp., with $750,000 of 6 per cent bonds and 363,000 shares of no par common stock.

Under the plan holders of the outstanding 50,000 shares of Edmunds & Jones stock would receive three shares in the new corporation and a bonus of $5 a share in cash for each share held. Holders of the 200,000 shares of Hall Lamp stock would exchange on a share for share basis plus $2.50 a share in cash. The Edmunds & Jones preferred stock would be retired.

The Edmunds & Jones Corp. is the largest manufacturer of automobile lamps in the United States and in Canada. It manufactures for both original equipment and for the replacement market and also is a large manufacturer of lamp parts and fittings. The Hall (C. M.) Lamp Co. ranks as one of the largest suppliers of lamps to the industry, making for both original equipment and replacement markets.

New York Takes Steps to Scrap Street-Car Lines

NEW YORK, June 7—Under a new State law providing that municipalities may acquire through purchase agreements trolley lines regarded as obstructing traffic, the Board of Estimate will hold a public hearing on June 22 on former Controller Craig's proposal that New York City buy three trolley lines operating in this city and replace them with buses. This is regarded as the first step by the city to make possible the substitution of bus service for a large proportion if not all of the surface trolleys now in operation.

C. I. T. Opens Berlin Office

NEW YORK, June 10—The establishment of a Berlin organization has been announced by Commercial Investment Trust. The German company will be known as the Commercial Investment Trust Aktien-Gesellschaft and will be under the management of Ludwig Lichenstadter.

Star Increases Discount

NEW YORK, June 10—Durant Motors, Inc., has increased its dealers' discount on Star cars effective June 1 according to announcement by Colin Campbell, vice-president.

Business in Brief

Written exclusively for* AUTOMOTIVE INDUSTRIES *by the Guaranty Trust Co., second largest bank in America.

NEW YORK, June 10—The more seasonable weather which prevailed during most of last week resulted in increased retail sales over much of the country and also served to stimulate the wholesale and jobbing trades, although the latter are beginning to be influenced by the seasonal ebb. The level of commodity prices declined during the week, while stock prices continued to advance.

CAR LOADINGS

Car loadings for the week ended May 22 again exceeded a million, totaling 1,039,385, as against 1,030,162 cars in the preceding week and 987,306 cars in the corresponding week a year ago. The net operating income of class 1 railroads during April amounted to $76,281,630. This represents an annual rate of return on property investment of 4.65 per cent, which compares with 4.14 per cent a year earlier.

BANK DEBITS

Bank debits to individual accounts reported to the Federal Reserve Board for the week ended June 2 were 1.3 per cent below the total for the preceding week and 2.3 per cent below that of a year ago.

FISHER'S INDEX

Fisher's index of wholesale commodity prices stood at 151.8 last week, as against 152.6 a week earlier and 151.2 four weeks earlier. Bradstreet's index for June 1 declined seven-tenths of 1 per cent from the index for May 1 and was 6.2 per cent below that of June 1, 1925.

FEDERAL RESERVE STATEMENT

Bills and securities held by the Federal Reserve banks increased $66,900,000 during the week ended June 2, with gains of $51,000,000 in discounts, $5,300,000 in open market purchases and $9,100,000 in Government securities. Note circulation rose $31,300,000 and deposits $18,100,000, while reserves decreased $29,000,000. The reserve ratio declined from 76.0 per cent to 74.3 per cent.

Loans of reporting member banks increased $88,000,000 during the week ended June 2. While loans secured by Government obligations and "all other" loans declined $12,000,000 and $14,000,000, respectively, loans secured by stocks and bonds rose $114,000,000. Investments increased $11,000,000, borrowing from Federal Reserve banks $49,000,000 and net demand deposits $15,000,000.

MONEY

Call loan rates ranged from 3½ to 5 per cent last week, as against 3½ to 4½ per cent a week earlier. Time loan rates were from 4 to 4¼ per cent, the same as the preceding week. Rates on commercial paper remained unchanged from 3¾ to 4 per cent.

Time Sales Mainly Sound, Says Reeves

Tells Dealers Automobile Trade is Favoring Best Practices —Replies to Critics

NEW YORK, June 9—Addressing the Brooklyn Motor Vehicle Dealers' Association luncheon today Alfred Reeves, general manager of the National Automobile Chamber of Commerce, said that instalment automobile selling on a sound basis with full consideration of "the three C's of credit—character, capacity to pay and capital of the buyer" provides a needed service to the public, supplies articles at lower cost, and helps to make the high standard of living that makes America what it is.

No one, he added, should condemn instalment buying because some of it is unsound. At least 90 per cent of automobile time sales were economically sound and the industry would gladly join with others in eliminating the 10 per cent that is unsound. The real problem was to embrace the good element and eliminate the bad, which was the program now being followed by dealers, manufacturers and finance companies.

With volume of sales in this country amounting to $30,000,000,000 annually, Mr. Reeves thought that the fact that $1,500,000,000 is outstanding on automobile paper did not seem alarming. With the Federal Reserve System, scores of business bureau and other modern methods of gathering business information there was small chance of precipitous drops in business that would suddenly throw large numbers of people out of employment.

Prices Lower for All Buyers

Increased and steady production gave even the time-buyer a lower price. On a $1500 car with $500 paid down there was added 8 per cent on the $1000 balance making $1080 payable in 12 months, covering all charges, including theft and fire insurance. If no sales were made on instalments that same car probably would cost $200 more because of the lower factory production.

Time sales, Mr. Reeves continued, "increase the cost only to the time buyer. The cash buyer enjoys a lower price as against the instalment buyer, whereas the average charge account gives one who pays his bill every 60 days the same price as the cash buyer, and sometimes gives him better service."

After citing other advantages Mr. Reeves as matters of caution urged that people should not buy higher priced articles than they can afford merely because they pay on instalments; that articles bought on time should be those which add to the buyer's earning power or to his permanent or semi-permanent assets, and which are necessary or very useful; and that down payments be sufficient to insure an equity for each buyer, large enough to encourage him to continue his payments; and that the time payments should never outrun the life of the article.

"The amount of instalment buying," said Mr. Reeves, "can always be limited by the bankers, who are justified in giving warning against abuses, thereby preserving the credit structure. There is no evidence that time selling has been a harmful factor in business during the past decade because savings in banks and in building and loan associations have almost doubled, life insurance in force has more than doubled, the building of homes has established new high figures, and the ownership of stocks and bonds by American citizens generally is best shown by the record number of stockholders in our railroads, public utilities and industrials.

Brunn New President of Body Association

DETROIT, June 9—Herman A. Brunn was elected new president of the Automobile Body Builders Association at the annual convention just closed here. Other officers elected were J. R. Way, first vice-president; A. V. Crump, second vice-president, and F. J. Radel, third vice-president.

Papers at the convention were by Harold Blanchard, of Motor, on, "What Does the Buying Public Want Today in Body Design?" Ray E. Plimpton, Bus Transportation, on "What Bus Operators Want in a Bus Body"; Merrill C. Horine, Mack trucks, on "Restrictions to Bus Design and Construction Resulting from State Regulations," and Harry C. Mougey, General Motors Corp., who described the development of pyroxylin finishes and summarized the present materials available and the methods used in their application. The discussion following the Mougey paper was led by W. J. Schlinger, E. I. duPont de Nemours & Co., Inc.; F. J. Pierce, Glidden Co., and L. Clayton Hill, Valentine & Co.

Federal Truck to Vote on Capitalization Change

DETROIT, June 10—On June 24, stockholders of the Federal Motor Truck Co. will vote on proposals to increase capitalization from 200,000 shares of $10 par value common to 500,000 shares of no par value and to pay a stock dividend of 100 per cent, which would use 400,000 shares of the new issue. The remaining 100,000 shares would be used for stock dividends, payable in quarterly installments of about 2½ per cent. An annual dividend rate of 80 cents a share on the new stock is recommended by the directors who will apply to the New York Stock Exchange to have the shares listed.

C. G. Declares Dividend

DETROIT, June 9—The regular quarterly dividend of 2 per cent on the preferred stock of C. G. Spring & Bumper Co. will be paid July 1 to stockholders of record June 24.

1925 Traffic Deaths Reached 22,500 Total

Bigger Increase Over 1924 Mark Than Had Been Expected Report Shows

NEW YORK, June 14—Automobile fatalities in the United States reached a total of 22,500 in 1925, an increase of about 2200 over the 1924 total, according to a survey made by the National Bureau of Casualty & Surety Underwriters for the statistics committee of the National Safety Council.

Although this analysis, based on official reports from 147 cities, indicates that the 1925 record was even worse than the National Safety Council's preliminary estimate of 21,000, the Bureau finds that safety education in the schools is having its effect and that certain cities which have done intensive safety work have secured marked decreases in automobile fatalities, proving that the situation is capable of control.

Automobile fatalities, excluding collisions with street cars and railway trains, increased from 17,600 in 1924 to 19,800 in 1925, and automobile grade crossing fatalities increased from 1688 to 1784.

The automobile death rate per 100,000 population increased from 14.9 in 1923 to 15.7 in 1924 and 17.2 in 1925. The increase from 1923 to 1924 was 5 per cent, and the increase from 1924 to 1925 was nearly 10 per cent. Automobile fatalities, according to the Bureau, are not only increasing in numbers but the rate of increase is also going up.

Fatality rates on the basis of registrations have been decreasing for several years, but the rate for 1925 was practically the same as that for 1924, with registrations increasing from 17,591,981 to 19,954,347.

Senate Acts on Road Bill

WASHINGTON, June 9—Agreement to take up the $165,000,000 Federal aid highway bill next Friday was made by the United States Senate on June 5, after exactly a month's delay and filibuster on the measure. The bill, which authorizes for the years 1927 and 1928, a continuance of the 50-50 Federal aid to States in highway construction, carries $87,500,000 for each of these years.

Graybar to Move Offices

NEW YORK, June 8—Graybar Electric Co., which on Jan. 1 succeeded the supply department of the Western Electric Co., has leased the entire fifteenth floor of the world's largest office building now under construction at Lexington Ave. and 43rd St., which will be known as the Graybar Building. The lease of approximately 40,000 square feet of space for the consolidation of its uptown offices calls for an aggregate payment of $2,000,000.

Hudson-Essex Cuts Prices on All Cars

New List on f. o. b. Basis Shows Reduction From Detroit a. y. d. Prices

DETROIT, June 10—The Hudson Motor Car Co. announced price reductions this week. The Essex coach is now $735, Hudson coach $1095, Hudson brougham $1395 and Hudson sedan $1550. The new prices are the lowest at which these cars have ever been sold. The reductions were made, says the company, at the end of the greatest five month's business it ever had, 114,500 cars having been shipped since January, or 17,000 more than in the same period of 1925. Retail sales for May aggregated 33,500. In April and May 68,500 cars were delivered to buyers.

The Hudson list is now quoted on an f.o.b. basis, and represents reductions from the Detroit delivered prices on the former a.y.d. basis of $30 on the Essex coach, $80 on the Hudson coach, $30 on the Hudson brougham, and $100 on the Hudson 7-passenger sedan.

Commenting on the new prices, Roy D. Chapin, chairman of the board said: "The reduced prices on Hudson and Essex cars are in line with the company's policy of offering outstanding values to the motoring public. This year to date has brought us an exceptionally active business. We have enjoyed too the advantage of large additions to our manufacturing facilities, permitting production of quality cars on a basis of great economy."

Suggest 108 Improvements in Patent Office Practise

WASHINGTON, June 9—A total of 108 recommendations for improvements in the U. S. Patent Office is contained in the report just made to Secretary of Commerce Hoover, by the special committee of 12 prominent industrialists and lawyers, on which was A. J. Brosseau, former member of the patents committee of the National Automobile Chamber of Commerce, and Milton Tibbetts, patent counsel of Packard Motor Car Co.

As a result of recommendations made by the Committee, hundreds of thousands of dollars and valuable time will be saved to patentees, of which the automobile industry ranks foremost in its business with the Patent Office.

Continues Canadian Plant

TORONTO, June 9—Dodge Brothers, Inc., Canadian factory, located here, will not close because of reductions in the automobile tariff, E. P. Clarkson, Canadian manager of the corporation announces. "We are carrying on business because we believe the Government will do the right thing," he declared. "We believe that the Government does not want to drive the motor industry out of Canada and that it will ultimately give such relief as is necessary for the continuance of the automobile manufacturing business in the Dominion." The company is not importing any cars from the United States plant and the latter factory, he said, could not take care of the Canadian demand.

Dodge Brothers Adds New Sport Touring

DETROIT, June 10—Following the recent introduction of a sport roadster, Dodge Brothers, Inc., has now added a 5-passenger sport touring listing at $880, the same price as the roadster.

The same general color scheme and upholstery as employed on the roadster is used on the touring car. Exterior finish is in pheasant green lacquer with a cream striping and natural wood finish wheels. Interior upholstery is in grey Spanish hand-buffed leather with the folding top in a heavy khaki duck.

Nickel trim is used extensively throughout. Bullet type headlights are standard equipment while disk wheels are optional at no extra cost. Additional equipment included in the price follows: cowl ventilator, automatic windshield cleaner, front and rear bumpers, rear view mirror, transmission lock and rear traffic signal.

Fisk Six Months Income Shows $87,332 Increase

CHICOPEE FALLS, MASS., June 5—Fisk Rubber Co. and subsidiaries report for six months ended April 30, 1926, net income of $2,124,593 after depreciation, interest and Federal taxes, equivalent after allowing for dividend requirements on the 7 per cent first preferred, 7 per cent first preferred convertible and 7 per cent second preferred stocks, to $1.60 a share earned on outstanding 811,827 shares of no par common stock. This compares with $2,037,261 or $1.68 a share on 797,892 shares of common outstanding in six months ended April 30, 1925.

Morse Chain Adds Building

ITHACA, N. Y., June 7—Morse Chain Co., plans a large addition to its plant here which will cost about $125,000 and have five stories with a floor area of 80 x 300 ft. The Morse Chain company is affiliated with the Thomas-Morse Aircraft Co. It is thought that aircraft will form a part of the manufacturing program with the completion of the addition, as well as the speeding up of the production of silent transmission chains.

Willys Builds "70" Coupe

TOLEDO, June 9—In addition to the two-door sedan listing at $1395 offered on the Willys-Knight "70" chassis, a two-passenger coupe at the same price as the new sedan has also been included in the line. With these two new body styles there are now four models offered on the smaller Knight-engined chassis by Willys-Overland, Inc.

Ford Profits Vary Patent Suit Shows

Range of 1 to 40 Per Cent at Periods Testified—Production Main Interest

DETROIT, June 10—Hearing of claims of the Parker Rust Proof Co. against Ford Motor Co., before William S. Sayres, master in chancery, covering certain damages and profits which the Parker company claims the Ford interests are accountable for, is still in progress. Several weeks more will probably be required before the hearing is concluded.

During the infringement period Ford company business totaled $4,222,325,856, it was testified, of which it is alleged over $11,715,605 was an infringement of the Parker company's patent rights. Profits due the Parker interests on this amount are computed at $2,663,988 for which it asks recovery. A claim for damages amounting to $392,799 and an alternate claim for profits of $1,256,676 is asked due to the saving and advantage to the Ford Motor Co. through the use of the patented process. During the infringement period it is claimed 67,158,256 lbs. of products were rust-proofed.

Profits, exclusive of interest on bank balances, bonds and investments and rent of buildings, etc., of the company from Jan. 1, 1917 to Nov. 12, 1924, aggregated $526,521,951. Sales for the same period totaled $4,384,357,826, figures introduced in evidence indicated.

Profits on various Ford automobiles has varied from less than 1 per cent on one type at a certain time to 40 per cent on another type on another time, Herbert L. Leister, chief auditor of the Ford Motor Co. testified. He also stated that during a period in 1924 the company took a loss of $231,000 on open cars. The company, he said, knew it faced a loss at the then prevailing price of open cars. Prices of Ford cars, he testified, is set at a mark to insure maximum production and it is charged to the organization to get the cost below that price.

"Mr. Ford himself does not pay much attention to costs," he declared, "when he begins to think that sales are dropping off and that something must be done to keep production at capacity and to keep 100,000 men working. He is not worried about a few thousand dollars as much as he is about keeping the men going and the public supplied with cars."

Detroit Deliveries Firm

DETROIT, June 7—Passenger car sales in Wayne county in May aggregated 10,422 as compared with 10,509 in April. May truck sales were 957 against 1078 in April.

Passenger car sales in the county for the first five months of the year aggregated 37,166, compared with 33,842 during the similar period in 1925. Truck sales for the same period totaled 4040 compared with 3049 in 1925.

GMC Backs $30,000,000 Plan for National Drivurself System

Yellow Truck & Coach Will Form Hertz Drivurself as Subsidiary to Buy Control of Present Local Companies and Nationalize Service

DETROIT, June 8—The Yellow Truck & Coach Mfg. Co., subsidiary of General Motors Corp., is organizing a new $30,000,000 transportation enterprise, John Hertz, chairman of the board announces. The capital stock of the company will be increased by $14,000,000 of which $10,000,000 is to be made immediately available for the new enterprise. It will mark the first time that General Motors interests have engaged in the business of operating transportation.

According to the announcement, the new transportation company will be known as the Hertz Drivurself Corp., and will act as a holding company, controlling state and local drivurself companies throughout the United States. Yellow Truck & Coach will purchase all the stock in the Hertz Drivurself Corp. out of the funds made available by the new stock issue. The General Motors Corp. will purchase such portion of the new issue at the price of $20 per share as is not taken up by the stockholders.

A special meeting of stockholders of Yellow Truck & Coach Manufacturing Co. will be held June 28, for the purpose of increasing the capital stock of the corporation and provide for the sale at this time of 700,000 additional shares of Class B stock. These shares will, in the first instance, be offered to the stockholders at $20 per share.

Officers and Directorate

Officers and directors of the Hertz Drivurself Corp. are: John Hertz, president; I. B. Babcock and M. L. Ross, vice-presidents; E. N. D'Ancona, secretary; executive committee, John Hertz, F. J. Fisher, J. L. Pratt, I. B. Babcock and M. L. Ross; directors, A. P. Sloan, Jr., John A. Ritchie, William Wrigley, Jr., Pierre S. duPont, John Hertz, J. R. Thompson, A. H. Swayne, F. J. Fisher, P. L. Emerson, J. L. Pratt, Donaldson Brown, C. A. McCulloch, J. J. Raskob, I. B. Babcock, E. N. D'Ancona, H. E. Foreman, L. S. Florsheim, O. W. Lehman, Robert Lehman, M. L. Ross and D. G. Arnstein.

Organization of the new company marks the first time in history that a transportation system has been offered to the public on a chain store plan of operation. Under the plan, it will be possible to secure a luxurious private car on a rental mileage basis in any city in the United States, use it either for local or touring purposes and leave it in any other city where the individual renting the car may happen to be when through with it.

"About two years ago I became convinced that there was a public demand for and a public service to be rendered by the establishment of a chain of operating companies throughout the United States which would be engaged in the business of renting high grade private automobiles without drivers on a mileage basis," declared Mr. Hertz, discussing the new project. "At that time there were already a large number of local concerns doing this class of business. I found, however, that it was being conducted in a more or less haphazard fashion, curiously resembling the taxicab business in its early stages and, unfortunately, in disrepute.

First Licensed Companies

"Sometime thereafter we organized the Hertz Drivurself system, a corporation capitalized at $1,000,000, all of the stock being owned by the Yellow Truck & Coach Mfg. Co. The object of this system was to license independent companies throughout the United States, and many of these companies are today operating under the name 'Yellow Drivurself System.' Within 15 months, we established in various cities approximately 300 stations engaged in the business of renting automobiles without drivers on a mileage basis. We now have about 6500 of these automobiles in service and have been entirely unable to keep pace with the demand.

"The drivurself business, or as it is sometimes called, the 'rent-a-car' business, is nothing more or less than the selling of a few dollars' worth of automotive transportation at retail on a mileage basis at any time when it happens to be needed. The customer takes the car, uses it as much or as little as he pleases and when he returns he pays for its use at so much per mile, as shown by the meter. This charge includes gasoline, oil and insurance. For the man who wants transportation, the difference between renting a car from the Hertz System and owning one is that, in the first instance, the individual purchases his automobile miles at retail without any of the responsibilities of ownership, and in the second instance, he purchases them wholesale, by buying the car.

Many States Now Organized

"The Hertz Drivurself Corp. has been organized in response to a great public demand. Its purpose is to make it possible to obtain the use of a Drivurself car anywhere in the United States at any time, except perhaps in the most remote rural districts. Not only will this be possible, but in large cities it is planned to have a sufficient number of stations so that they will not be over one mile apart. Already we have fairly completed operations covering the states of California, Florida, Illinois, Kentucky, Massachusettes, New Jersey, New York, Ohio, Oregon, Washington, Wisconsin and British Columbia. These operations will be completed within the next three or four months. Surveys have been made of every other state in the Union and the work of organizing these states as part of the system is already under way.

May Supply Commercial Vehicles

"I look forward to the time when commercial vehicles of all kinds can be supplied on this same basis of rental at so much per mile. With our complete chain system once established throughout the United States, it will be possible for us to furnish commercial vehicles on a mileage basis which will make their use much more economical for many establishments than the ownership and maintainance of trucks."

It has also been announced that the Hertz Drivurself Corp. is prepared to extend financial aid to existing local companies engaged in the business of renting cars without drivers for the purposes of expansion and also to bring them into the Hertz national and state systems. There are at the present time thousands of such companies which, because of limited capital in a purely localized business are unable either to supply the demand or extend any service to their customers beyond their own communities. By bringing them into this great national system, patrons of these local companies will be able to secure national service though continuing to patronize their home industry, according to the plan.

This method of organizing will give to the local companies the national standing and prestige due to the backing of the General Motors Corp. together with the organization, management and experience of the Hertz transportation interests. Local and state Drivurself companies, however, will be in the first instance under a board of directors which will include local representation with a minority interest in the stock of these companies held locally. The headquarters of the Hertz Drivurself Corp. will be in Chicago, but all the state and local companies will have local headquarters.

$200,000 Plant for Denby

LOS ANGELES, June 9—The Denby Motor Truck Co. of California, a subsidiary of the Denby Motor Truck Co. of Detroit, has announced plans for the erection of a $200,000 manufacturing and assembly plant in the central manufacturing district, Los Angeles, to serve the Pacific Coast and handle exports to Australia, South America and the Orient.

Fulton Company Builds

MILWAUKEE, June 8—New construction costing about $50,000 has been undertaken by the Fulton Co., manufacturer of automotive equipment and specialties, at its main works in West Allis, suburb of Milwaukee.

Paris-Berlin Planes Open Daily Service

Commercial Lines Connecting Marseilles and Other Points Also are Inaugurated

PARIS, May 29 (*by mail*)—Commercial airplane lines from Paris to Marseilles and from Paris to Berlin were inaugurated this week and will operate daily, carrying passengers, mails and freight. On the Paris to Marseilles line, distance 477 miles, the planes stop at Lyons, thus giving air connection with Geneva. The Paris-Marseilles line is also linked up with the Paris-London line, thus making it possible to travel from London to Marseilles in 10 hours. By train and boat the journey occupies 48 hours.

The Paris to Marseilles line is operated by the Air Union (Farman company) and the initial trip was made by five Spad-33 cabin planes equipped with a Salmson 260 hp. engine.

The distance from Paris to Berlin is covered in 7½ hours, this including a landing at Cologne and another stop at Essen. At present the journey by train occupies 20 to 23 hours. When in full working order, the Paris-Berlin service will be linked up with the existing lines to Hamburg and Copenhagen.

The Air Union is operating the Paris to Berlin line in conjunction with Luft Hansa company. The French planes will be Farman machines fitted with two Farman engines, but later these will be replaced by giant planes built specially for this line. The first German plane to enter France since the war was a three-engine Junkers.

After maintaining an air mail service from Toulouse to the West Coast of Africa for the last three years, the Latecoere company is now prepared to carry passengers to Rabat, Casablanca, and the Senegal. There will be a service each way four times a week between these important centers.

A daily flying boat service from Trieste to Turin is now in operation. At Turin the machines alight on the River Po, close to the Fiat automobile factory. A flying boat service is also operating from Genoa to Palermo, Sicily, making the trip in seven hours, compared with 24 hours by the best train and boat connections.

Dirigibles Craft of Future

WASHINGTON, June 6—Giant dirigibles rather than airplanes will constitute the principal transportation medium of the future, it was predicted this week by Starr Trescott, of the Bureau of Aeronautics, Navy Department, in a speech before a group of scientists. He declared that the recent air flights over the North Pole had heightened interest in aviation and envisioned in the minds of the public transcontinental air routes for the future.

TAKING "FLYER" AIDS TURKISH AVIATION

WASHINGTON, June 8 — A novel scheme for raising funds to provide for the construction and maintenance of an aviation program for Turkey, has been reported to the government by Trade Commissioner Gillespie at Constantinople.

The plan provides for the sale of 50,000 lottery tickets each month. The first sale was in April, and yielded the government approximately $150,000, leaving a similar amount to be distributed as prizes. First prize to the lucky ticket holder will be $50,000, second $20,000 with seven smaller prizes. The scheme is designed to yield the government annually between $1,500,000 to $2,000,000—all to be expended in behalf of aviation.

Buick Sells 205,182 in First 10 Months

FLINT, June 7—A total of 205,182 new Buicks have been delivered to owners since the present line of Buicks were introduced last August, Buick Motor Co. reports. In the first 10 months after the introduction of the 1925 models 144,776 were sold, setting a record. The 1926 record, however, is higher by 60,406 cars, or 41.7 per cent.

The sale of 205,182 Buicks of the latest model in 10 months exceeds by many thousands the sale of any previous model in an entire 12 months' period, it was said.

Demand for Buicks continues at the high level reached in April, when 31,632 cars were delivered to owners. In May, 29,966 deliveries were made. Deliveries for the first five months since Jan. 1, total 110,026 cars. This exceeds the record for the same period in 1925 when 78,612 cars were delivered up to the first of June.

Every shipment and delivery record of the Buick Motor Co. has been broken time after time since the 1926 line was introduced, according to E. T. Strong, general sales manager. Deliveries for the last few months have been running ahead of production from 7000 to 10,000 cars and there is every indication of this continuing.

Franklin May Sales High

SYRACUSE, June 5—Only twice in 25 years has Franklin's May business exceeded the volume attained last month, according to a statement issued by the Franklin Automobile Co.

During the first five months of 1926, Franklin sedan business represented 68 per cent of the company's total, which is a gain for this type of car over 1925. Last year the sedan made up 64 per cent of the total, according to factory executives, who state that the three-passenger, clear-vision coupe ranks next to the sedan.

New Oakland Plant to Make 1600 Daily

Will Have Capacity for 1000 Pontiacs and 600 Oaklands for 1927 Market

PONTIAC, June 7—Oakland Motor Car Co. factories here will be practically doubled as a result of an enlargement program just announced by A. R. Glancy, president. The extensions will increase the total investment in the company from $25,000,000 to $50,000,000. Decision in the matter was reached at a meeting of the executive committee of the General Motors Corp. here, June 1.

Work will be started at once on a complete new automobile factory to cost $5,000,000 and to be devoted exclusively to the manufacture of the Pontiac. It will have a daily capacity of 1000 units.

Erection of the new factory will make the present buildings which embrace two complete automobile plants under one roof, now used for both Oakland and Pontiac production, available exclusively for Oakland production. It will have a daily capacity of 600 cars.

The Oakland expansion will call for concurrent expansion of the present Fisher body plant at Pontiac, and also other plants supplying Oakland with axles, steering gears, transmissions, etc.

According to present plans the new Pontiac factory will be ready for production by Jan. 1, 1927. It will give the Oakland Motor Car Co. an annual capacity for 400,000 units.

More than $3,500,000 has been invested in Oakland factories during the past year, to increase production of both Oakland and Pontiac Sixes. This, together with the newly announced expansion program, approximates an investment of nearly $9,000,000 within little more than a year and will represent an increase in plant capacity from 250 to 1600 cars a day.

During the past 10 months more than 1500 new dealers have contracted to sell Oakland and Pontiac automobiles, and, since January 1, new dealer agreements have been written at the rate of nearly 200 a month.

Aircraft Bill Provides for 1000 Plane Minimum

WASHINGTON, June 7—The purchase of 1947 new airplanes and funds for the maintenance of a minimum of 1000 heavier-than-air crafts, is provided in a bill passed this week by the Senate. Altogether the measure appropriates $85,000,000 to be expended over a five-year building program for the naval air service. The measure has already passed the House. But sixteen minutes were required to debate the measure and pass it in the Senate.

Provision is also made for the erection of two rigid airships, each three times the size of the wrecked Shenandoah and to cost not more than $5,000,000.

U. S. Plants Build 87% of World's Cars

Automotive Division Figures for 1925 Show One Point Recession From 1924

WASHINGTON, June 10—The United States accounted for 87 per cent of the 4,763,428 automobiles manufactured during 1925 in the six leading motor vehicle producing countries of the world according to completed revised figures just released here by the Automotive Division of the Department of Commerce. This country accounted for 4,154,269 passenger cars while the other five countries—Canada, France, Italy, the United Kingdom and Germany—produced the remaining 13 per cent.

During 1925, the revised figures further show, the United States accounted for but 61 per cent of the combined exports of these six countries, which amounted to 498,427 automobiles.

Citing 1924 production figures of automobiles by the nations comprising the "big six" which were 3,972,681 of which the United States accounted for 88 per cent, officials of the Automotive Division point to the fact that the slight decrease in 1925 as compared with 1924 "indicates the necessity for American manufacturers to take full cognizance of increasing European competition in certain markets even if it is not yet manifest that a proportionately greater number of European cars are being exported. In 1924 United States automobile exports were 55 per cent of the combined leading countries, shipments amounting to 320,390 passenger and freight automobiles."

It is made plain that the foregoing study deals with passenger cars. "The comparisons," the statement accompanying the statistics sets forth, "includes all automobiles, and necessarily so, because of the lack of European production statistics in which the various classes of automobiles are separated."

Set Palmetto Hearing Date

WASHINGTON, June 10—Hearings before the United States Supreme Court on the automobile insurance suits brought by the various state commissioners of insurance against the Chrysler Sales Corporation and the Palmetto Fire Insurance Company have been set for the week of October 11.

The right of the Chrysler corporation to insure the automobiles it sells to customers will be argued in five different suits which have been combined, however, as one assigned case.

Autocar Sales Increase

PHILADELPHIA, June 8—Autocar Co. reports a 117 per cent increase in sales of its 3-ton power dump truck in the first quarter of the year, which the company says is due in large part to increased road building activities in many parts of the country.

EVANSVILLE OFFERS TRACT TO GRAHAMS

EVANSVILLE, IND., June 9—A group of Evansville business men, members of the Evansville Chamber of Commerce, and backed by that body with an offer of a substantial tract of land, are endeavoring to influence the Graham brothers to again choose Evansville as the site for their business operations if the brothers decide to reenter the industrial field.

A letter of invitation has been addressed to Joseph B. Graham, president of the Hercules corporation, builder of motor truck bodies, and former head of the Graham Brothers motor truck industry Mr. Graham is expected here in a few days, at which time a Chamber of Commerce delegation plan to present their proposal and offer inspection of the site.

Motor Regrinders Rename Association

NEW YORK, June 7—Returning from the recent third annual convention held in Indianapolis, R. C. McWane, secretary, said today that the National Motor Regrinders & Rebuilders Association had changed its name to Associated Automotive Engine Rebuilders as more expressive of its nature and purposes.

The 14 local associations also will use this new name but will be regarded as sections of the national association.

The convention developed considerable discussion of the increasing tendency of engine rebuilders to become engine parts jobbers.

E. P. Chalfant, executive vice-president of the National Standard Parts Association, discussed cooperation among automotive trade associations, and other papers were by T. H. Hill, of the Norton Co., on cylinder and wheel regrinding; Joseph Pflum, of Heald Machine Co., on past, present and future of regrinding, and H. W. Knapp, of McQuay-Norris Mfg. Co., on merchandising automobile engine rebuilding parts.

These officers were elected: F. W. Young, Eagle Machine Co., Indianapolis, president; H. W. Lahmann, Buckeye Regrinding Co., Cincinnati, vice-president; W. A. Baker, Will A. Baker Co., Chicago, treasurer, and R. C. McWane was reelected secretary.

McGill Makes Bearings

VALPARAISO, IND., June 8—McGill Metal Co. is increasing production steadily of its new Schubert ball-bearings, following their successful introduction to the trade. The bearing is named for F. R. Schubert, vice-president of the company, formerly with U. S. and Strom. T. L. Robinson has joined the McGill company as quality manager in charge of metallurgy, heat treating and inspection.

Panama Concession Stirs Senate Action

Would Have Government Investigate Grants to British Company Near Canal Zone

WASHINGTON, June 10—Chairman Borah, of the Senate Foreign Relations Committee, alarmed by reports that British interests have obtained concessions on vast tracts of public lands in Panama on both sides of the Canal Zone, this week introduced a resolution calling on the State and War Departments for all available information on the subject.

The British concern is known as the Panama Corporation, Ltd., and is controlled by capital interested in oil and rubber development. The grant includes some 2,000,000 acres of land that American experts have classified as suitable for rubber cultivation. The territory consists of Panama public land.

It is said that while the concession itself, as a business venture, could hardly be termed as violating the Monroe Doctrine, nevertheless if the British have been given authority, as reported, to "police" the territory, the State Department may be compelled to take cognizance of it.

Says Tract Unfit for Rubber

NEW YORK, June 7—Alexander Lyle-Samuel, a member of the original group which obtained the concession, says the Panama Corp., Ltd., concerning whose concession in Panama Senator Borah seeks information from the State and War Departments, is not interested in rubber growing or military bases in Panama. The corporation is interested only in mining, he adds, and the British government is in no way connected with it.

In reply to the Congressional suspicion that the concession was asked for in order to grow rubber, Mr. Lyle-Samuel insists that the corporation has no interest in rubber and that any rubber expert would know that rubber cannot be grown on the concession.

Seek Tarnishing Cause

WASHINGTON, June 9—Tests to determine the cause and remedy of tarnishment of automobile headlamp reflectors will be undertaken by the U. S. Bureau of Standards, following a conference held in Cleveland last week on the subject. The tests will be made under the supervision of Dr. G. W. Vinal, chief of the Electrical Chemistry Section.

Willys "70" Orders Increase

TOLEDO, June 10—Willys Overland Co. reports unfilled orders on its model "70" Willys-Knight as in excess of 5000. Total production on all models is now at the rate of 1500 daily. Expansions are being made to its main factory buildings here and at Pontiac and Elmira to permit of increased schedules.

Men of the Industry and What They Are Doing

P. W. Litchfield Named to Rubber Association Board

Paul W. Litchfield, president of the Goodyear Tire & Rubber Co. has been elected a director of the Rubber Association of America, succeeding to the vacancy created by the death of G. M. Stadelman, formerly director and first vice-president of the association, and former president of Goodyear. C. B. Seger, chairman of the board and president of United States Rubber Co., was elected first vice-president of the association and Mr. Litchfield was chosen to succeed Mr. Seger as second vice-president. Resolutions of regret on the death of Mr. Stadelman were adopted.

Pinto Holley District Head

E. S. Pinto has been appointed district sales manager in charge of the central western district of the United States for the Holley Carburetor Co., according to L. D. Burnell, secretary-treasurer of the company. He will have charge of Holley sales in Minnesota, Wisconsin, Michigan, Illinois, Indiana, Ohio, Kentucky and Tennessee. Headquarters office will be maintained in the Arcade Building, Chicago.

Managers Named Directors

Charles M. Gearing, manager of the Meriden division, and Sidney E. Stockwell, manager of the Hartford division of the New Departure Mfg. Co., have been elected directors of the company. Both men have risen from the ranks and have earned their new honors through achievement covering a long period of company relationship.

F. E. Moskovics to Sail

Mr. and Mrs. F. E. Moskovics will sail for Europe June 18 on a six weeks' business trip. With Mr. and Mrs. L. L. Corum, who are now in England, Mr. and Mrs. Moskovics will make a swing around the continent in a Stutz sedan, returning early in August to Indianapolis.

Warner Officers Elected

A. P. Warner has been elected president of the Warner Patterson Co., Chicago, at the annual meeting just held. Frank F. Soule is vice-president and J. H. Cattell, general manager, secretary and treasurer.

Henry Ewald Sails

Henry T. Ewald, president of Campbell-Ewald Co., Detroit, has sailed on the Rotterdam with Mrs. Ewald and their two children, for a three months' European trip.

Jolls With New Britain

E. L. Jolls, formerly with Packard Motor Car Co. and the Marmon Motor Car Co., has joined the sales force of the New Britain Machine Co. and will be located in Detroit.

EXPORT OFFICERS NAMED IN DETROIT

The Export Club, of Detroit, at its annual meeting, elected W. E. Morrison of the Paige-Detroit Motor Car Co., president.

Other officers are Samuel Fitzpatrick, first vice-president; E. Porter West, second vice-president; George H. Zimmerman, treasurer; Howard H. Tewkesbury, secretary, and R. R. Daley, George H. Howell, Percy Owen, J. Howard Rees, F. B. Sides and H. B. Trix, directors.

Dedicate Dort Memorial

FLINT, June 7—Members of the Flint Country Club recently dedicated a beautiful memorial flag staff, erected at the club in the memory of the late J. Dallas Dort, automobile manufacturer, who dropped dead there a year ago, while playing golf.

The base of the staff is of cut stone while the pole is set in a massive bronze casting bearing plaques in memory to Mr. Dort.

Join Randall Organization

F. E. Ransier ad R. D. Hughes have become connected with the Fred M. Randall Co., Detroit, the former as production manager and the latter as a copy staff member. Mr. Hughes was formerly in the advertising department of Packard Motor Car Co. and Chrysler Sales Corp. Mr. Ransier was in the engraving business.

Stamm With Miller-Aschheim

A. C. Stamm has been appointed general manager of the Miller-Aschheim Co., Inc., of New York, and will also take charge of the engineering department. Mr. Stamm was formerly connected with the Raybestos Company.

Jordan Resigns from Ford

J. D. Jordan has resigned as assistant sales manager of the Ford Motor Co. He has been in charge of sales of Fordson tractors and Ford trucks for the past several years. No successor has been appointed.

Middleton in Detroit

Mortimer Middleton has been appointed Detroit representative for Walden-Worcester, Inc., with headquarters in the General Motors Building. Mr. Mortimer was formerly connected with S. K. F. Industries.

Burnish With A. O. Smith

Howard J. Burnish, formerly with H. H. Franklin Mfg. Co. and later chief engineer of the gas engine works of Worthington Pump & Machinery Corp. and works manager of the Milwaukee Motor Products, Inc., is now with the A. O. Smith Corp.

Mooney Sees Export Boom Waiting Equitable Prices

J. D. Mooney, president, General Motors Export Co., who recently returned from a European trip, left last night in company with other executives of the company, on a trip to General Motors plants and will return to New York about June 18.

Mr. Mooney spent four months surveying conditions in England, France, Spain and the Near East. He believes that if these countries could obtain automobiles at the cost at which they are being made and operated in the United States the expansion of automobile business abroad would be enormous.

G. M. Export Men Change

M. F. Lawrence, manager of the general manufacturing department of the company, has returned to New York after completing a tour of the European plants and operations. G. K. Howard, who has spent the past year and a half at the export operations in Copenhagen and Antwerp, has returned to New York to make preparations for his departure to the Far East where he will become regional director for that territory.

A. J. Wieland, formerly managing director of General Motors International A/S at Copenhagen, is now in charge of operations in the Far East and will be stationed at Alexandria, Egypt. Morris Clark, who has been managing director of the Paris office, has been named regional assistant for Europe, assuming the duties of Paul Hughes, who has been assigned to the staff of the European regional director. P. Bayard has been appointed sales manager of General Motors, France. E. E. Kaiser, formerly on the staff of regional director for Europe, has been made assistant managing director of General Motors Peninsular S/A with headquarters at Madrid.

Davies in New Zealand

Glynn Davies, vice-president of the General Motors Acceptance Corp., on completion of the work of organizing the General Motors Acceptance Corp. for General Motors New Zealand Ltd., on which he is now engaged, will undertake the organization of a similar branch for the new operation in Australia. He is now in the Antipodes where he will remain until the completion of this work.

E. E. Main Sells Interests

E. E. Main, president of the U. S. Bearings Co., Inc., Indianapolis, has disposed of his financial and active interests in the company and after a short vacation will reenter the replacement parts field.

Treslar Now in Chicago

Glen Treslar, manager of the Buffalo office of The Van Dorn Electric Tool Co., has been transferred in a similar capacity to the Chicago office.

Ford Uses Lacquer on Part of Output

DETROIT, June 7—Ford Motor Co. dealers here have received the Tudor sedan and coupe models finished in a special green lacquer. The shade of this finish resembles the regular colors on the present closed models except that it has a dull satin-like effect. Black enamel finish is retained on the open models.

It is understood less than half of the closed-car production is coming through with the new lacquer finish, and a 100 per cent schedule will not come into effect until officials of the company see how the public likes the new finish.

The lacquer finish for the closed cars was developed in the Ford company's own experimental laboratories.

Valley Succeeds Huffman, Builds Two New Trucks

ELKHART, IND., June 7—Two new truck models are now in production by the Valley Motor Truck Co., of Elkhart, Ind., successor to the Huffman Brothers Motor Co. The new models are the EH with a carrying capacity of 3000 pounds, listing at $1890, and the BC with a carrying capacity of 2½ tons listing at $2295.

The model EH uses a 4-cylinder Hercules OX engine with 4x5 bore and stroke, rating 25.6 hp., and developing 45 hp. at 2200 r.p.m. The model BC uses a 4-cylinder Continental S-4 engine with 4 x 4½ bore and stroke, rating 28.9 hp., and developing 50 hp. at 2200 r.p.m.

Standard wheelbase is 132 in. on the EH model and 140 in. on the BC model, but longer chassis may be had in either model at additional cost. Pneumatic tires 33x5 are furnished on the lighter model and solid tires 34x4 in front and 34x7 in rear are on the heavy model.

The personnel of the new company manufacturing Huffman trucks is headed by Louis Poncher, president and general manager. Other officers are B. E. Smith, vice-president; S. T. Smith, secretary, and L. C. Smith, treasurer. M. H. Hoshaw is in charge of the sales department and R. E. Chiddister of the engineering department.

North Carolina Registers Mostly Low-Priced Cars

RALEIGH, N. C., June 7—During May a total of $614,398 was collected by the state in automobile licenses, gasoline, and automobile title taxes, swelling the total amount collected from these for the first eleven months of the present fiscal year to $11,668,392. This is more than $1,500,000 in excess of the amount collected from the same sources during the entire preceding fiscal year when collections totalled $10,130,486.

Collections for the eleven months are classified as follows: Licenses, $5,298,084, gasoline $6,230,919 and titles $139,387. Automobile and truck licenses issued number 391,038. For automobiles alone 335,920 licenses were issued. The figures show that the great majority of these cars are of the lower priced makes. License fee is based on horsepower starting at $12.50. Cars calling for this license charge number 302,038. Department officials expect that licenses sold during the new year will number 475,000.

Northwest Sales Gain 50%

SEATTLE, WASH., June 5—Latest figures on new car sales show big increases this year in the state of Washington, compared to 1925. The total for the first quarter this year is 9923, compared to 6381 a year ago.

Used car sales are more brisk than a year ago, indicating a healthy condition in automotive business in the Northwest. Much sales effort is now being put forth to sell the used car, and sales are getting to be a real factor in the business of the average dealer. Much advertising is devoted to put over sales and some of the concerns are beginning to make a profit in this department.

Ethyl Distribution to Expand Gradually

NEW YORK, June 5—Arrangements for expanding the distribution of ethyl gas for sale in combination with gasoline throughout the United States are now being made by the Ethyl Gas Co.

Ethyl gasoline is now being sold by those distributors which were under contract with the Ethyl Gas Co. for the sale of ethyl gas before sales were suspended pending the investigation by U. S. Surgeon General's department, the report of which was favorable to the sale of this gas.

So far there is only one tetraethyl plant, and within a week or so production of one unit of this plant will be started. For the time being, distribution will be limited to capacity of this plant.

Ethyl gasoline will be sold in different sections under the names of the distributors in those sections, but in practically all cases the words "with ethyl" will be added to the name of the gasoline advertised by the distributor. For example, the Standard Oil Co. of Kentucky, is already selling such gasoline under the name "Red Crown gasoline with ethyl."

An extensive advertising campaign is under way but at the offices of the Ethyl Gas Co. today the report that upwards of $4,000,000 would be spent in this campaign in the next two years was described as exaggerated. It was explained that the company's appropriation for advertising was being divided among its distributors who in turn combined it with their own appropriations for advertising in their territories.

Churchville Expands Plant

PHILADELPHIA, June 9—The Churchville Mfg. & Laboratories Co. has expanded its enterprise by installing a brass foundry at its plant. The addition, according to President V. S. Fry, will aid materially in rendering better service to the established Wharton distributing organization.

Developments of the Week in Leading Motor Stocks

NEW YORK, June 10—There was a better underlying tone to the motor stocks during the past week and several issues, which are known to be in a strong trade position or in which special developments of a financial nature are pending, broke away from the general group, scoring sensational advances.

Willys-Overland common was an outstanding feature among the low priced issues, touching new high ground on the movement at above 27 and showing an advance of around 10 points from its recent low price. The movement in this stock is based on expectations of a large increase in the company's business and earnings. A strong pool with good inside connections is understood to be active in the market.

General Motors common was another strong spot in the motor list, advancing to above 130 the price which it reached before the $4 extra dividend was declared. As $5.75 was deducted when the stock sold ex-dividend, the corresponding quotation is now above 136. Company is understood to have earned over $13 a share in the first five months of the year, a total which exceeds earnings in any previous full year prior to 1925.

Pierce-Arrow fulfilled predictions that directors would resume dividends on the 8 per cent preferred stock by last week declaring a quarterly dividend of $2 a share on the issue. The resumption of preferred dividends is an important event in Pierce-Arrow's affairs, emphasizing the full recovery of this company from the post-war depression. While no action toward liquidating the arrears has been taken, it is understood that tentative plans are being considered. Some action similar to that taken by Willys-Overland in paying off accumulated dividends with an issue of common stock is not unlikely.

Other motor stocks failed to make any material headway during the week, while Hudson and Studebaker, in plentiful supply, closed the week at slightly lower prices.

Rubber shares gave indications of increasing activity and strength. Practically all those issues showed gains of one to two or more points. Goodrich sold above 50, Fisk above 19 and U. S. Rubber above 57.

There was less activity in truck stocks. Profit taking in Mack Trucks brought about some recession from last week's level. White Motors and Yellow Truck & Coach eased off in sympathy.

The accessory issues showed few significant changes.—E. S.

N.A.C.C. to Insure Truck Facts Heard

Committee Will Assemble All Data for Hearings—Study New Insurance

NEW YORK, June 5—The directors of the National Automobile Chamber of Commerce are appointing a committee of five or six truck manufacturing members to provide ways and means to insure that all the facts about motor vehicle common carriers, especially trucks, are presented at all congressional hearings on the Cummins bill and other measures designed for Federal regulation of buses and trucks.

This subject was discussed to a considerable extent by the truck members at their meeting this week and various views were presented as to the proper means for providing truck manufacturing representation in the hearings.

The truck members authorized the Chamber's insurance committee to confer with officials of the Underwriters' Laboratories in an attempt to reach an understanding regarding the method by which the underwriters are now offering a 10 per cent discount in fire insurance rates on commercial cars which have met certain conditions.

On Jan. 1, 1924, the Underwriters' Laboratories completed the working out in considerable detail of an examination schedule for determining the fire hazard on commercial cars. Only recently the Chamber insurance committee was notified by the Underwriters' Laboratories that a 10 per cent discount from standard fire insurance rates would be granted on trucks which had undergone an examination in accordance with this schedule of specifications. The examination is made by personal inspection of the model in the factory.

Freight Damage Losses Reduced to $1.60 a Car

NEW YORK, June 7—Freight damage losses have been reduced 60 per cent by railroads in the past five years, according to the report of W. E. Metzger, chairman of the traffic committee of the National Automobile Chamber of Commerce. Since 1921 loss and damage claims on all commodities were reduced from $96,730,376 to $38,772,097 in 1925. Claims charged to automobiles, trucks and accessories were reduced over the same period from $3,311,560 to $1,331,702.

In 1921 the freight claims were paid on a total wholesale value of automotive shipments of $2,212,068,420. In 1925 the lesser claim was paid on a wholesale valuation increased to $4,210,174,963. Carloads of automotive commodities shipped in 1921 totaled 359,472 and in 1925 totaled 832,713. The claims paid per car in 1921 were $9.21 and in 1925 were $1.60.

HUDSON SALVAGES SHIPPING MATERIAL

DETROIT, June 7—The Hudson Motor Car Co. is now operating a large lumber salvage plant to replace an older one. From 5000 to 6000 lbs. of nails are recovered every day. These come from boxes and boards which have been used in shipping various materials to the plant. The principal salvage product, however, is the lumber, which averages from 75,000 to 100,000 feet a week. Many of the nails are in suitable condition for re-use, while the balance are sold for scrap junk.

Many boxes can be re-used while the rest are sawed up so that all worth-while lumber is saved. The scrap material left over is fed into a huge grinder which reduces the balance to fragments about the size of shelled corn. This is fed by compressed air into one of the furnaces of the main power plant, and, at capacity, this material supplies 800 to 1000 horsepower an hour.

Study of Transport Needs Aids Truck and Bus Growth

NEW YORK, June 5—Credit for the large increase in the use of trucks and buses during the past year belongs to scientific study, in the opinion of Edward F. Loomis, secretary of the Motor Truck Committee of the National Automobile Chamber of Commerce, speaking before the transportation classes of the University of Pennsylvania.

There has been a great advance in the study of the motor vehicle's relation to transport during the past year or two, Mr. Loomis pointed out. Shippers and trucking companies are studying routes so as to use less congested streets and thereby reduce time losses. Scientific planning of streets and highways is adding to the facilities of motor travel.

Electric railway companies and independent companies installing bus lines are making studies both of existing volume of population and probable trends. All these factors, the speaker stated, point to continued development of motor traffic on a sound basis.

$10 Day Demurrage Upheld

WASHINGTON, June 7—The legality of a $10 per day demurrage charge on freight shipments was upheld by the U. S. Supreme Court this week, settling a five years' test case.

Substance of the decision is that "all demurrage charges have a double purpose. One is to secure compensation for the use of the car and the track it occupies. The other is to promote car efficiency by providing a deterrent against undue detention by the shipper, either before or after loading."

Declare Overbuying Instalment Danger

Method of Listing Obligations of Buyers Necessary, Says Credit Association

NEW YORK, June 4—The National Association of Credit Men in analyzing a recent survey among credit executives issues a statement calling attention to "the dangers to business that accompany consumers' overbuying and the permitting of individuals to mortgage their income too far into the future for the security of credit."

These are described as the "principal evil factors that reside in instalment buying," and other future danger spots in instalment merchandising are described as "a detrimental effect on social conditions and on general trade, a general laxity in accepting the credit of individuals and the encouragement of the production of luxuries to the injury of non-luxury goods."

More than 70 per cent of the credit executives in manufacturing, wholesaling and banking, whose opinions were tabulated, criticized instalment merchandising, as now conducted, on the basis of consumers' overbuying. These executives, many of whom are treasurers, owners and important officials in their concerns, stressed the need of an effective syste mto discover consumers' overbuying on instalments so that credit risks could be properly judged.

This 70 per cent agree that instalment selling encourages or permits extravagance, because credit managers as a rule have no efficient way of determining when an individual has purchased up to his capacity as the wholesale credit man has in determining the outstanding indebtedness of a retail merchant or jobber.

Tariff Proposal Upsets Truck Market in England

NEW YORK, June 7—While Parliament has not yet approved the Chancellor of the Exchequer's recommendation that the McKenna duties of 33 1/3 per cent be placed on imported commercial cars and trucks in addition to passenger cars, the British automobile situation is serious for American and Continental truck exporters, according to John D. Biggers, managing director of Dodge Brothers (Britain), Ltd.

Mr. Biggers arrived here on the Berengaria and will discuss the British situation with Dodge Brothers officials in Detroit, when a definite policy in reference to the British situation will probably be worked out.

While the tariff situation is in a state of flux, Mr. Biggers believes that business conditions in England will improve constantly. During the long period of only fair business he adds that the British have built up a sound foundation for all industries.

Steel Buying Light as Half-year Nears

Good Sized Contracts for Automotive Steels in Third Quarter Reported

NEW YORK, June 10—While current automotive demand for steel is light, some good-sized sheet and strip-steel contracts are reported to have been placed for third-quarter shipment by leading Detroit interests, and others are negotiating for their July and August requirements. Parts makers are replenishing their bar stock to the extent to which they are making shipments to their own customers.

Automotive alloy-steel producers show increased eagerness for orders, and in some instances quotations $1 or $2 a ton below recent prices are made. Judging from talk in automotive circles, considerable misapprehension prevails with reference to the possible effect on the nickel market of the use of chromium alloys for plating. There are those who imagine that chromium-plating will much diminish the demand for nickel. As a matter of fact, approximately only 2 per cent of the total nickel consumption is used for plating, so that even though chromium-plating should cut into the demand for nickel by platers, the loss would hardly be felt.

The "big" end of nickel consumption is in 3½ per cent automotive alloy steels. three-quarters to four-fifths of the total nickel consumed goes into steel as an alloy, and the steady increase in demand for that purpose many times offsets any possible competition by chromium as a plating material. Technically important, therefore, as the development of chromium plating may be to the automotive industries, marketwise it is a negligible factor.

Full-finished automobile sheets are easy at $4.20, Pittsburgh, but it is generally thought that the sheet market, as a whole, is scraping bottom, and that, even though a further shrinkage in the demand take place, prices will not yield much further. The leading interest's unfilled tonnage statement for May 29, to be published today, is not expected to put a different face on the steel market. There is the usual wide gap between guesses as to the extent of the decline in the Corporation's backlog, but no one looks for any surprises.

Pig Iron—The $1 per ton decline in the price of foundry iron, valley basis, that has been recorded in the last five weeks emphasizes the complete control of the pig iron situation by heavy tonnage buyers. Automotive foundries do not seem over eager to profit from this latest dip in price, and continue to buy from hand to mouth.

Aluminum—News from Italy is to the effect that the Milan Montecatini Co. plans construction of an aluminum plant at Cotrone, close to Istrain bauxite deposits. While in itself of little importance from the American consumer's point of view, this news shows the tendency to increase foreign production, and only through such increased production can more genuine competition in the American market be hoped for. Domestic market unchanged.

Copper—Apparently an understanding has been reached with reference to the export combine, and improved export conditions are certain to be reflected in fractional advances at home. Domestic demand is of a routine character.

Tin—The market is easy for futures, but spot metal is not over-plentiful.

Lead—Much better demand from storage battery makers features the situation.

Industry Increases Hardwood Purchases

ATLANTA, GA., June 1—With excellent weather having apparently resulted in an increased volume of new car business all over the country, hardwood sales to the automobile industries during the last four or five days of May recovered the volume shown earlier in the season, with orders placed on a basis that would indicate the manufacturers confident of the future.

Several sizeable orders were booked during the period for future requirements, for the first time in five or six weeks, while the volume of orders for current needs showed steady expansion. The inquiry is also much larger, and would indicate that larger manufacturers do not have any surplus hardwood stock.

Trade with implement manufacturers, however, is still declining, only a comparatively few orders being booked for immediate needs. Ash prices are the same, but maple prices have declined $2 to $4 per thousand feet since May 25.

Federal Reserve Report Shows Car Stocks Lower

CHICAGO, June 8—New car wholesale sales in the seventh Federal Reserve district increased 20.6 per cent in April over April, 1925, with a value increase of 13.9. Retail sales showed a gain of 4 per cent. New car stocks decreased 22.9 per cent from March, though 78.1 per cent higher than in April, 1925. Used car sales were 11.9 per cent more than in March, though slightly under sales in April, 1925. Used car stocks decreased 3.1 per cent from March but were 40.9 per cent higher than in April, 1925.

Rubber Arrivals Lower

NEW YORK, June 7—Crude rubber arrivals in the United States during May were about 32,000 tons, compared with 35,004 in April, 42,677 in March and 32,867 in Rebruary, according to Henderson, Helm & Co., Inc.

It is estimated that May shipments from the East to the United States were about 30,200 tons, compared with 26,200 in April, so that June arrivals will be considerably under those of the following month.

Financial Notes

Vauxhall Motors, Ltd., a British subsidiary of General Motors Corp., for the year ended Dec. 31, 1925, reports profit of £36,083 after expenses and debenture interest, comparing with £50,067 in 1924. There was appropriated £4,500 for preference dividend and £23,018 for additional expenses, leaving balance of £8,570, which with £25,941 brought in made total of £34,511 carried forward.

General Motors Corp.—J. P. Morgan & Co. are selling privately a block of $9,000,000 of General Motors preferred stock at 118½, it was announced yesterday. There will be no public offering. The sale does not involve any new financing for the corporation and will not increase the total of preferred stock outstanding.

Kelsey Wheel Co. has declared regular quarterly dividend of $1.50 on the common, payable July 1 to stock of record June 18. New directors elected were Bert Morley, succeeding Charles W. Fox, and Malcolm G. Campbell, who takes the place of Joseph N. Barton, both resigned.

Stromberg Carburetor Co. of America, Inc.—For the quarter ended March 31, 1926, reports net profit of $134,497, after expenses, Federal taxes, etc., equivalent to $1.68 a share earned on 80,000 shares of no par stock. This compares with $160,305 or $2 a share in the first quarter of 1925.

American Bosch Magneto Corp. has called for redemption the Gray & Davis 7 per cent bond issue of 1932 to be effected August 1 at $105 and interest. Redemption funds are being deposited with the New York Trust Co. and the First National Bank of Boston.

Reo Motor Car Co.—An extra quarterly dividend of 1 per cent, in addition to regular quarterly dividend of 2 per cent, has been declared, both payable July 1 to stock of record June 15. The par value of the stock is $10.

Commercial Investment Trust has declared quarterly dividends of 90 cents on the common, $1.62½ on the 6½ per cent preferred, and $1.75 on the 7 per cent preferred, all payable July 1 to stockholders of record June 15.

Hudson Motor Car Co.—New York Stock Exchange committee on securities has ruled that the capital stock of the Hudson Motor Car Co. be not quoted ex. the 20 per cent stock dividend on June 10 and not until June 17.

Mack Trucks, Inc., has declared dividends of $1.50 on the common, $1.75 on the first preferred and $1.75 on the second preferred, all payable June 30 to stockholders of record June 15.

Chysler Sales Corp. directors have approved the employees' preferred stock purchase plan whereby employees are permitted to subscribe at $100 a share on deferred payments.

Delco-Light Sales High

DAYTON, June 8—Retail sales of Delco-Light Co. aggregated $11,250,000 in May as compared with $4,300,000 in May, 1925. A total business of $80,000,000 in 1926 is indicated, according to E. G. Biechler, president and general manager.

Pierce-Arrow Back on Dividend Basis

First Payment on Preferred Since 1921 Shows Return to Strong Position

NEW YORK, June 8—Declaration by directors of Pierce-Arrow Motor Car Co. of dividend of 2 per cent on the $10,000,000 accumulated preferred stock means the resumption of current quarterly dividends, which have not been paid since April 1, 1921. Accumulated unpaid dividends on this stock amount to $40 per share or a total of $4,000,000 on the 100,000 shares outstanding.

Resumption of preferred dividend payments is the natural result of satisfactory and constantly increasing business during the last few years. Myron E. Forbes, president, adds that the success of the Pierce-Arrow series 80, with the constantly increasing volume of truck and bus business and the regular output of the large passenger car, has resulted in steadily increasing earnings.

The company has no bank loans outstanding and its bank balance is $2,582,740 compared with $2,099,805 at the end of 1925. Net income last year was $1,629,781 compared with $751,060 in 1924. Net income in the first quarter this year was $328,982 against $200,416 in the 1925 quarter.

Managers Securities Corp. Earns $4,002,307 in 1925

NEW YORK, June 10—The annual report of the Managers Securities Co., the organization of General Motors executives, shows the amount earned in 1925 under its contract with General Motors Corp. as $4,002,307 after Federal tax. Dividends of $480,000 during the year left a balance of $3,522,307 credited to its class A surplus.

Dividends received from the corporation and interest earned aggregated $6,903,176, which after payment of dividends on the preferred stock of the Managers company and other deductions, left an accrual of $5,325,449 credited to its class B surplus.

During the year $8,280,000 of 7 per cent preferred stock was retired, leaving outstanding obligations reduced to $732,446 for current taxes and $15,690,000 in remaining preferred. Following all disbursements during the year, cash on hand was $13,608.65.

C. I. T. Assets $81,177,081

NEW YORK, June 8—The financial statement of Commercial Investment Trust Corp. as of Dec. 31, 1925, shows total current assests of $81,177,081 and total current liabilities of $40,431,264. Current assets include cash of $19,690,752, receivables and secured loans of $61,486,329. Current liabilities include notes payable of $38,599,261, and accounts payable of $1,748,669. Surplus is shown as $4,539,083.

WAUKESHA DROPS CLEVELAND SUIT

CLEVELAND, June 7—Dismissal of the suit of Waukesha Motor Co. against the Cleveland Automobile Co. was asked and obtained in Federal Court here by counsel for the Waukesha Company, as the Cleveland Automobile Company has discontinued use of the Ricardo head since the suit was initiated. Further action was regarded by officials of the Waukesha Company as being unessential.

The suit as initiated was to restrain the Cleveland company from using the Ricardo design in its engine manufacture.

May Rim Production Shows Slight Drop

CLEVELAND, June 10—Rim production figures for May totaled 2,178,947 as compared with 2,538,412 for May, 1925, and with 2,284,171 in April this year according to report of the Tire & Rim Association of America, Inc. For the five months to May 31 production in 1926 was 11,085,542 as compared with 11,409,551 in the same period last year. Production totals for May in principal sizes were as follows:

Size	May 1926	May 1925
Clincher:		
30 x 3½	240,593	423,155
Balloon:		
28 x 3½	980,903	943,459
28 x 4	241,147	270,715
29 x 4	313,661	160,934
30 x 4½	105,423	113,074
30 x 5	32,197	43,321
31 x 5	39,813	32,388
High Press.:		
30 x 3½	9,088	54,784
32 x 4	9,722	32,704
32 x 4½	22,813	64,873
Truck "20":		
30 x 5	66,188	76,258
Truck "24":		
36 x 6	7,876	8,406

Gasoline Output Reaches Highest Record in April

WASHINGTON, June 5—Gasoline production in the United States during the month of April broke all previous records, according to announcement just made here by the U. S. Bureau of Mines. Production was 987,633,000 gallons, representing a daily production of 32,921,000 gallons, which was 5 per cent more than the daily production in March and 13 per cent gain over April, 1925.

Consumption during April was 831,410,000 gallons, a demand of 10 per cent over March consumption. Stocks of gasoline on hand May 1 amounted to 1,926,725,000 gallons, or 64 days supply based on the average automotive consumption.

To Probe Gasoline Prices

WASHINGTON, June 5—The Trammell resolution, authorizing a Federal investigation into gasoline prices, was adopted this week by the United States Senate, without a record vote.

Stutz Five Months Shows Profit Gain

May Company's Largest Month —New Car Charged to 1925 Operations

INDIANAPOLIS, June 8—Shipments by Stutz Motor Car Co. of America, Inc., in the first five months of the year were in excess of 2250 cars with a net operating profit of approximately $340,000, according to F. E. Moskovics, president. May was the leading month for the company to date with shipments totaling 700 cars and profits of about $225,000, the largest monthly total ever achieved by an Indianapolis manufacturer. Losses were incurred in the first two months of the year owing to heavy expense incidental to starting production of the new vertical "8" line.

Commenting on the outlook for the company, Mr. Moskovics said:

"There is a very healthy demand for the company's product as evidenced by the fact that unfilled orders indicate an assurance of operation at full capacity for some time to come. The present product is original in design and noncompetitive in character, and we believe that the active sales, with resultant profits are not liable to be disturbed either by competition or unfavorable business conditions for some time to come."

For the year ended Dec. 31, 1925, the company reports net loss of $1,660,385, which compares with net loss of $517,567 in 1924. Net sales for 1925 totaled $2,420,336 as against $2,347,887 in 1924. The heavy loss in 1925 was ascribed to the charge against operations of $1,000,318 which included the entire development and engineering expense on the new car introduced in 1926; obsolesence of inventories and equipment used in manufacture of discontinued models; losses on old purchase commitments, and reserves to provide for losses to be sustained in liquidating branches.

The balance sheet as of Dec. 31, 1925, shows total assets of $5,579,247, and profit and loss surplus of $2,277,620.

Industrial Acceptance Shows Business Increase

NEW YORK, June 8—Industrial Acceptance Corp. shows an increase of 84.39 per cent in the volume of Studebaker dealers' paper handled during the first five months of 1926, compared with the same period in 1925, according to a statement issued by Arthur J. Morris, president of Industrial Acceptance.

Attributing this increase to the lowered interest rates to the automobile buying public which are now in effect, Mr. Morris said: "All indications point to continuation of this growing volume. Those who criticise the buyers of automobiles on the deferred payment plan should distinguish between the proper use of credit and its improper abuse."

Underwriters Drop Theft-Lock Credits

NEW YORK, June 5—The uncertainty which has prevailed among car makers and car lock manufacturers in regard to the future insurance status of cars equipped with coincidental locks has been dispelled by a decision of the National Automobile Underwriters Conference to eliminate all discounts and credits for anti-theft locks of all kinds.

In writing insurance policies next year under the new plan the car owner will get no discounts because of locks on his car but the rate will be fixed according to the theft experience data of the underwriters for his make of car. Whether a lock shall be provided and, if so, what kind, will be left to the judgment of the manufacturer whose interest will naturally be to furnish with the car whatever device will tend to produce the best theft experience and thus secure low rates on his product.

This new plan takes effect when new insurance rating books are published. Usually this occurs about the first of each year but it is possible that books for 1927 will be published somewhat earlier than usual.

Netherlands Proposes Tax

WASHINGTON, June 7—The Netherlands has been invaded by the good roads movement with the result that the government is proposing a road improvement project to be covered by an annual tax on motor vehicles. The projected tax is from $2.41 to $5.02 on passenger cars, $4.02 on trucks, $5.03 on auto buses, and $4.02 for motorcycles.

Coming Feature Issue of Chilton Class Journal Publication

Sept. 30—Automotive Industries Annual Production Issue

Balloon Tires Favored in Danish Tire Market

WASHINGTON, June 7—American tire manufacturers will do well to hold their own in the profitable Danish market according to a dispatch received by the Rubber Division of the Department of Commerce from Commercial Attache H. Sorensen at Copenhagen. "European manufacturers," he reports, "can be depended upon to spare no effort to retain their strong position."

The increasing number of automobiles in Denmark has resulted in increased imports of tires. On Jan. 1, 1926, the estimated number of motor vehicles in operation included 45,500 passenger cars, 820 motor buses, 14,000 trucks, and 18,800 motorcycles. Approximately 50 per cent of the replacement demand is for inch-clincher tires; from 20 to 25 per cent for metrick clinchers; and from 25 to 30 per cent for straight sides. The current trend is increasingly toward balloon tires.

Fesco Plants Expanded

CLEVELAND, June 7—Ferbert-Schorndorfer Co., manufacturer of Fesco paints, has started work on a new office building and will also add a new manufacturing building, the fourth in five years, to its main plant.

Assign Subjects for Battery Convention

NEW YORK, June 7—At the annual meeting of the National Battery Manufacturers Association at Roosevelt Hotel, New York City, June 25 and 26, J. B. Perlman of the Hartford Battery Co., will relate "The Romance of the Battery," A. F. Strayer of the U. S. Light & Heat Corp., will discuss refused sight draft shipments and there will be a presentation by the Harvard Economic Service.

This is only part of the program, the subjects for which speakers are now being obtained including the outlook for radio batteries, grids and oxide mud, health in the battery plant and the success of the trickle charger and eliminator during the past year.

O. B. Towne, commissioner of the association, announces that on June 7 the offices of the association and the Asbestos Brake Lining Association will be moved from 17 West 42nd Street to Room 1009, at 578 Madison Avenue.

Inter-City Air Line Ready

WASHINGTON, June 6—The establishment of a regular airplane passenger service between Washington and Philadelphia, was tentatively announced here this week by Secretary of Commerce Hoover. The plans provide for three planes, of 12 passenger capacity each, to ply between the two cities. Contracts, it was stated, have already been let for the building of the planes—of the Fokker type. No announcement as to who will run the planes or operate the line was made. It is understood here that the line is being financed by Philadelphia capitalists.

Calendar of Coming Events

SHOWS

Brussels .....Dec.

Buenos Aires .....Dec. 7-20
Ninth Argentine Automobile Show, Palermo Park.

Chicago .....Nov. 8-13
Coliseum, Automotive Equipment Association.

Chicago .....Nov. 15-19
Hotel Sherman, National Standard Parts Association.

Chicago .....Jan. 29-Feb. 5
National, Coliseum, National Automobile Chamber of Commerce.

Cleveland .....Oct. 4-8
Public Auditorium and Annex, American Electric Railway Association.

Geneva .....June 10-20
Third Annual Automobile, Motor Cycle and Cycle Exposition.

Lima .....July 28
First Peruvian Automobile Show, Under Auspices Peruvian Touring Club.

London .....Oct. 4-9
Olympia Motor Cycle.

London .....Oct. 21-30

Milan .....Sept.
Fifth International Road Congress and Exposition.

New Haven .....Sept. 7-10
Machine Tool Exhibition.

New York City .....Sept. 13-18
Madison Square Garden, Radio Manufacturers Show Association.

New York .....Jan. 8-15
National, Grand Central Palace, National Automobile Chamber of Commerce.

Paris .....Oct. 7-17
Auto Salon, Grand Palais.

Prague .....Sept.

CONVENTIONS

American Electric Railway Association, Public Auditorium and Annex, Cleveland .....Oct. 4-8

Associated Manufacturers of Fabric Auto Equipment, Inc., La Salle Hotel, Chicago .....Nov. 13

Automotive Equipment Association, Mount Royal Hotel, Montreal, Canada .....June 14-19

Automotive Equipment Association, Coliseum, Chicago .....Nov. 8-13

National Association of Automobile Show and Association Managers, Drake Hotel, Chicago .....July 27-28

National Automobile Chamber of Commerce, Factory Service Managers, Book-Cadillac, Detroit.....June 15-16

National Battery Manufacturers Association, Roosevelt Hotel, New York City .....June 25-26

National Standard Parts Association, Hotel Sherman, Chicago.....Nov. 15-19

National Tire Dealers Association, Inc., Memphis, Tenn. .....Nov. 16-18

Society of Industrial Engineers, Bellevue-Stratford Hotel, Philadelphia..June 16-18

S. A. E. MEETINGS

National

Boston, Nov. 16-18, National Transportation and Service.

Chicago, Sept. 21-23, Production Engineering, Hotel Sherman.

Philadelphia, Sept. 2-3, Aeronautical.

RACES

Altoona .....Sept. 6

Atlantic City .....July 17

Atlantic City .....Sept. 25

Charlotte, N. C. .....Aug. 23

Laurel, Md. .....Oct. 23

Le Mans, France .....June 12-13
Rudge-Whitworth 24-hour stock car race.

Los Angeles .....Nov. 25

Marseilles .....June 27
French Grand Prix, Miramas Track.

Salem, N. H. .....July 5

Salem, N. H. .....Oct. 12